A COMPARISON OF A SPLIT AND COMBINED PLANT ON A 5000 TON SURFACE EFFECT SHIP (SES)

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by

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ABSTRACT

A methodology based on range is used to evaluate the effectiveness of a split and combined propulsion/lift system on a 5000 ton surface effect ship (SES) with a length to beam ratio of 6.5. The vessel operates in the 45-50 knot regime, is propelled by partially submerged supercavitating controllable pitch marine propellers, and uses second generation gas turbines as the prime movers. The equations for calculating drag and the method used to select the marine propeller, to design the lift fans, to design a transmission system, and to perform weight estimates for several plant components are summarized. The combined plant, which utilizes the same prime movers for both the lift and propulsion function requires a heavier and more complex transmission system than the split plant which separates the lift and propulsion function. However, the results show that the combined plant achieves a greater range than the split plant due to the fuel economies gained from operating both the lift and propulsion systems with the same prime mover.

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TABLE OF CONTENTS

LIST OF FIGURES

LIST OF FIGURES (continued)

LIST OF TABLES

LIST OF TABLES (continued)

LIST OF TABLES (continued)

SYMBOLS AND TERMS

TERM	DESCRIPTION	UNITS
A	Overall fan diameter	ft
A_D	Cross sectional area of duct	ft^2
B_c	Cushion beam	ft
BG(n)	Bevel gear (number)	
b_p	Number of planet gears (number of torque paths)	
B_{SH}	Sidehull beam	ft
C	Centerline distance between a single reduction gear and its pinion	in
C_{DO}	Aerodynamic drag coefficient	
C_{DS}	Seal drag coefficient	
C_f	ITTC friction factor	
C_r	Sidehull wavemaking drag coefficient	
D	Fan impeller diameter	ft
d	Overall gear diameter	in
D_A	Aerodynamic drag	lb
D_C	Air leakage or discharge coefficient	
d_g	Gear pitch diameter	in
$D_i(d_i)$	Inside shaft diameter	in
D_m	Propulsor mounting drag	lb
$D_o(d_o)$	Outside shaft diameter	in
D_P	Propeller diameter	ft
d_p	Planet (pinion) pitch diameter	in
d_r	Ring gear diameter	in

TERM	DESCRIPTION	UNITS
D_R	Ram or momentum drag	lb
D_S	Seal drag	lb
d_s	Sun gear pitch diameter	in
D_{SF}	Specific fan diameter	
D_{SFD}	Sidehull form drag	lb
D_{SHF}	Sidehull frictional drag	lb
D_{SHW}	Sidehull wavemaking drag	lb
$(D_{TOT})_{RW}$	Total craft rough water drag	lb
$(D_{TOT})_{SW}$	Total craft smooth water drag	lb
D_{WM}	Cushion wavemaking drag	lb
E	Modulus of elasticity	lb/in^2
F	Face width	in
f'	(Fuel Rate)/(Design Fuel Rate)	
F_B	Bouyant force of sidehulls	lb
f_L	Newman - Poole wave drag parameter	
F_L	Froude number	
f_{ss}	First critical speed	Hertz
G	Geometry factor	
g	Acceleration due to gravity	$32.2\ ft\text{-}lbm/lbf\text{-}sec^2$
H	Fan discharge head rise	ft
h	Average draft	ft
h_b	Water depression due to cushion pressure	ft
$(HP)_C$	Cushion horsepower required	H.P.
H_g	Gearbox height	in

TERM	DESCRIPTION	UNITS
h_g	Cushion air gap	ft
$(HP)_{BP}$	Propeller horsepower available	H.P.
$(HP)_{BR}=(HP)_P$	Required propeller horsepower	H.P.
h_w	Average wave height	ft
I	Moment of inertia	in^4
J	Polar moment of inertia	in^4
J_P	Advance coefficient	
K	K-factor (gear surface durability factor)	lb/in^2
K_Q	Torque coefficient	
K_T	Thrust coefficient	
L	Overall fan length	ft
l	Gearbox length	in
L_c	Cushion length	ft
L_{CS}	Critical shaft length	in
L_D	Duct length	ft
L_S	Maximum wetted sidehull length	ft
M_g	Overall reduction ratio	
M_{g1}	First reduction ratio	
N	Revolutions per minute	RPM
n	Revolutions per second	RPS
n'	RPM/Design RPM	
n_f	Number of fans	
n_p	Speed of rotation, input pinion	RPM
NPC	Net propulsive coefficient	

TERM	DESCRIPTION	UNITS
N_s	Specific fan speed	
n_s	Speed of rotation, sun gear	RPM
P	Power	H.P.
P_c	Cushion pressure	lb/ft^2
PGF	Planetary gear, fan	
PGP	Planetary gear, propulsion	
Q	Air volume flow rate	ft^3/sec
q_a	Air dynamic pressure	lb/ft^2
Q_D	Q-factor (size factor)	H.P./RPM
Q_f	Volume flow rate of air per fan	ft^3/sec
Q_p	Propeller torque	ft-lb
Q_{SL}	Volume flow rate of air in the static lift case	ft^3/sec
q_w	Dynamic water pressure	lb/ft^2
Q_{WP}	Volume flow rate of air in the wave pumping case	ft^3/sec
R	Size factor	H.P./RPM
R_e	Reynolds number	
RIGF	Reversing idler gear, fans	
RIGP	Reversing idler gear, propulsion	
S_c	Cushion area	ft^2
S_{CR}	Critical shear stress	lb/in^2
SFC	Specific fuel consumption	lbs/H.P.-HR
S_{Fn}	Reference frontal area	ft^2
S_g	Cushion discharge area	ft^2

TERM	DESCRIPTION	UNITS
SF(n)	Shafting between fans (number)	
SI(n)	Interconnecting shafting (number)	
$(SA)_n$	Surface area	ft^2
SP(n)	Shafting, propulsion section (number)	
S_s	Shear stress	lb/in^2
SS-n	Sea-state, number	
ST(n)	Shafting between turbines (number)	
T	Thrust	lb
t'	Torque/Design Torque	
U_t	Fan tip speed	ft/sec
V	Ship velocity	ft/sec
V_a	Speed of advance	ft/sec
V_r	Relative wind velocity	ft/sec
V_w	Headwind velocity	ft/sec
W	Displacement	lb
$\overline{W}$	Tangential gear tooth load	lb
W_B	Bevel gear weight	lb
W_C	Shaft coupling weight	lb
W_f	Fan weight	lb
W_{FS}	Weight of fuel carried by the split plant	tons
W_{FB}	Weight of fuel burned off	lb
W_{FC}	Weight of fuel carried by the combined plant	tons
w_g	Gearbox width	in

TERM	DESCRIPTION	UNITS
W_{HC}	Weight of the combined plant lift and propulsion hardware	tons
W_{HS}	Weight of the split plant lift and propulsion hardware	tons
W_M	Miscellaneous weight	lb
W_P	Weight of planetary reduction gears	lb
W_{PR}	Weight of a titanium propeller	lb
W_S	Shaft weight	lb
w_s	Single reduction gear weight	lb
W_{SC}	Ship weight supported by cushion air pressure	lb
W_{TC}	Total weight of the combined plant	tons
W_{TS}	Total weight of the split plant	tons
ρ_a	Mass density of air	$0.0024(\text{lb-sec}^2)/\text{ft}^2$
ρ_s	Density of steel	$0.28\ \text{lb/in}^3$
ρ_w	Density of water	$1.99\ \text{slugs/ft}^3$
ν_w	Kinematic viscocity of sea water @ 59°F	$1.28\text{X}10^{-5}\ \text{ft}^2/\text{sec}$
ω	Specific weight	lb/ft^2
ψ	Head coefficient	
μ	Poisson's ratio	
ξ	Diameter ratio	
Δ_{FL}	Full load displacement	tons
γ_d	Duct efficiency	
γ_f	Fan efficiency	
γ_H	Hull efficiency	

TERM	DESCRIPTION	UNITS
γ_M	Mounting efficiency	
γ_{max}	Maximum fan efficiency	
γ_o	Propeller open water efficiency	
γ_R	Relative rotative efficiency	
γ_T	Transmission efficiency	
γ_{TF}	Fan transmission efficiency	

1. INTRODUCTION

The term "surface effect ship" refers to a vessel partially or completely supported by air pressure. In its modern context a surface effect ship (SES) consists of two rigid sidewalls which penetrate the free surface and two flexible end seals which follow the wave motion. The rigid sidewalls and flexible seals contain the cushion air pressure. While a small part of the total lift is provided by sidewall bouyancy, most of the lift force is obtained from the cushion air pressure. Lift fans are used to pressurize the cushion and compensate for air leakage which occurs through the forward and after seals. The general configuration of an SES is shown in Figure 1.

Since the SES is largely supported by an air cushion, its resistance is significantly lower than a conventional displacement ship. The lowered resistance means the SES is capable of much higher speeds in smooth water and mild sea states than conventional ships. Figure 2 illustrates the characteristic difference in drag powering requirements in smooth water between a conventional ship and two SES's of the same displacement. The EHP for the SES's in Figure 2 includes both lift and propulsion requirements. The total resistance of an SES is the sum of wavemaking drag, sidewall wavemaking drag, sidewall frictional drag, sidewall form drag aerodynamic drag, momentum drag, seal drag and appendage drag. The vessel parameters which affect these drag components are cushion pressure, cushion length, cushion beam, and frontal area. One of the most important parameters which affects the drag of an SES is the ratio of cushion length to cushion beam (L_c/B_c). Low L_c/B_c SES's are generally more suitable for

operation at speeds greater than 50 knots. The U.S. Navy prototypes, SES-100A and SES-100B are low L_c/B_c designs (1.95 and 2.00 respectively) as is the proposed 3000 ton SES. These ships were designed for speeds in excess of 70 knots. High L_c/B_c SES's appear to have more application at lower speeds. Figure 2 also illustrates the variation of EHP with speed for various L_c/B_c ratios. The speed below which the high L_c/B_c SES has an advantage over the low L_c/B_c SES is about 60 knots. As discussed in Reference 1, these high L_c/B_c vessels become attractive for open ocean operation in sizes of approximately 5000-6000 tons.

Three types of propulsors have been proposed for SES's: waterjet, air propeller, and marine propellers. Waterjet systems have the lowest propulsive efficiency of the three propulsors. They also have the added disadvantage of the large weight penalty of the water carried in the system. The advantages of waterjet propulsors include simpler, lighter transmissions due to the high rotational speeds and considerable flexibility in the location of prime movers. The SES-100A and the proposed 3000 ton SES are both waterjet designs.

Air propulsion has been successfully applied to a variety of fairly modest sized commercial and military vehicles. The types of air propulsors include free and shrouded air propellers, turbojets and turbofans, cruise fan and Q-fan systems, and centrifugal and axial flow fan systems. Air propulsors have good low speed performance and offer high potential efficiency as ship speed increases. Most air propulsors require a large deck area and suffer from high radiated noise levels.

Marine propeller systems consist of a fixed or controllable reversible pitch (CRP) propeller, a propeller mounting appendage, a transmission with

reduction gears, and one or more engines. Since subcavitating propellers suffer excessive cavitation damage at speeds much above 40 knots, supercavitating and superventilated propellers are usually considered for application with SES designs. Supercavitating propellers can operate either fully or partially submerged. Fully submerged propellers can use either pod-strut or inclined shaft mountings. Either pusher (propeller aft) or tractor (propeller forward) pod mountings can be used. Partially submerged propellers are usually transom mounted. With gas turbines, CRP propellers are usually installed to avoid complex reversing gears. The SES-100B is powered by supercavitating variable pitch, partially submerged propellers and has achieved speeds in excess of 80 knots. The primary advantage of the partially submerged propeller configuration is the low appendage drag. Problems with marine propeller propulsion systems include fluctuating propeller forces and the complications which arise from right angle transmission systems.

An SES must also have installed a lift system consisting of fans which pressurize the air cushion. The propulsion and lift system can either be combined or split. The combined plant utilizes a single power source to provide power both to the propulsion system and lift system. The split plant separates the lift and propulsion functions. One power source is dedicated to the lift system and another is dedicated solely to the propulsion system.

The purpose of this paper is to compare the attributes of a split and combined plant on a 5000 ton SES. The propulsors will consist of partially submerged supercavitating controllable reversible pitch propellers. The power source for both the split and combined plants will be gas turbines.

The cushion length to cushion beam ratio (L_c/B_c) investigated will be 6.5. The operational performance characteristics selected for the vehicle being studied are: a top speed of 50 knots in sea state 1 (SS-1), and a top speed of about 45 knots in SS-3. These performance characteristics would meet or exceed those of existing combatants or ships planned for the 1980's and are consistent with the projected SES state-of-the-art. A summary of the other characteristic dimensions of the SES in this study is provided in Table 1. These dimensions are depicted graphically in Figure 3.

2. THE METHODOLOGY USED IN COMPARING THE SPLIT AND COMBINED PLANTS

In order to compare the split and combined plants it is necessary to establish a common basis for measuring the merit of the two systems. The common measure of merit used will be vehicle range. This section describes the methodology used to arrive at this value.

The methodology must account for the effect of the decrease in the ship's weight as the fuel is consumed. This effect is very important for an SES for two reasons:

1. A substantial percentage of the gross weight of an SES is devoted to fuel (the fuel fraction for the SES-100B is 47% of the gross weight).
2. A significant amount of power is required for the lift function (for the high L_c/B_c SES of Figure 2 for instance, approximately 20% of the total power is devoted to lift at 50 knots).

Several simplifying assumptions are necessary initially in order to proceed with the methodology. These assumptions are summarized in Table 2 which also indicates whether they apply to the split plant, the combined plant, or both plant types. The first step in the analysis, Chapter 3, determines the total propulsive horsepower required to satisfy two different operating characteristics, i.e. 50 knots in a calm sea and 45 knots in a sea state 3. In Chapters 4 and 5 respectively, a compatible propulsor and lift system is selected. These components are not optimized, rather, effort is made to insure the feasibility of those components selected. The selection of the propulsor and lift fans, in turn, influences the selection of the

engines for the split and for the combined plants (Chapter 6). After the engines are selected, the transmission system for the two plants is designed (Chapter 7) and the total propulsion and lift system weights are estimated (Chapter 8). These weights will include the following components and elements of the propulsion and lift system:

1.0 Propulsion System

1.1 Gas turbines (including accessories normally supplied with the engines)

1.2 Gas turbine inlet and exhaust ducting

1.3 Propulsors

1.4 Propulsor shafting

1.5 Propulsion transmission system

1.5.1 Reduction gears

1.5.2 Transmission shafts

1.5.3 Bearings and supports

1.5.4 Clutches

2.0 Lift System

2.1 Gas turbines

2.2 Gas turbine inlet and exhaust ducts

2.3 Lift fans

2.4 Lift fan ducting (including the forward and after seal ducts)

2.5 Lift system transmission

2.5.1 Reduction gears

2.5.2 Transmission shafts

2.5.3 Bearing and supports

2.5.4 Clutches

The value of fuel fraction was arbitrarily selected for this analysis. The proposed 3000 ton LSES will have a fuel fraction of approximately 40%, but an operational SES will have a lower fuel fraction therefore a fuel fraction of 25% was selected. With a 5000 ton SES this fuel fraction equates to 1250 tons of fuel. The split plant will be used as a base. That is, the total weight of propulsion and lift hardware plus fuel for both plants will be limited to the total weight of the split plant. This means that if the combined plant has a lower hardware weight it will carry more fuel and vice versa. The resulting weight equations are:

$$W_{TC} = W_{FC} + W_{HC} \tag{2.1}$$

$$W_{TS} = W_{FS} + W_{HS} \tag{2.2}$$

$$W_{TS} = W_{TC} \tag{2.3}$$

where W_{TC} = total weight of the combined plant

W_{FC} = weight of fuel carried by the combined plant

W_{HC} = weight of combined plant lift and propulsion hardware

W_{TS} = total weight of the split plant

W_{FS} = weight of fuel carried by the split plant

W_{HS} = weight of the split plant lift and propulsion hardware

Starting with 100% fuel aboard, the lift power required for the full weight condition is determined in Chapter 9. Using the sum of the lift and propulsion power, the SFC and fuel flow rate for the combined plant at the full power level will be calculated. This same procedure is followed for the split plant case. The SES is then operated at a constant speed until

the fuel aboard has been reduced by 20%. The distance the SES has traveled during this time interval is calculated. These calculations are repeated at the 80%, 60%, 40%, 20%, and 0% fuel levels. The sum of the distances traveled at each fuel level represents the integrated range of each SES. This concept is illustrated in Figure 4.

The combined and split plants are then compared to determine which system provides the greatest range.

3. DRAG CALCULATIONS

The first step in the analysis is the determination of the propulsive power requirements for a 5000 ton SES with the characteristics and performance requirements described in Table 1. In order to calculate the propulsive horsepower requirements, the total craft drag must be known. The drag of an SES is made up of hydrodynamic components, aerodynamic components, and drag components unique to these vehicles, i.e. seal drag. The total craft drag is composed of the following elements:

a. Cushion wavemaking drag

b. Sidehull wavemaking drag

c. Sidehull frictional drag

d. Sidehull form drag

e. Aerodynamic drag

f. Ram or momentum drag

g. Seal drag

h. Appendage drag

Cushion wavemaking drag will be examined first. This drag component is the result of the pressurized cushion moving over the water and generating waves. Work at the David Taylor Naval Ship Research and Development Center (DTNRSDC), Reference 2, has resulted in the graph of Figure 5. This graph relates the non-dimensional wave drag parameter f_L to Froude number F_L for various L_c/B_c ratios. The cushion wavemaking drag can be calculated by:

$$D_{WM} = \frac{4\, f_L\, P_c\, W}{\rho_w\, g\, L_c} \qquad (3.1)$$

where D_{WM} = cushion wavemaking drag, lbs

f_L = Newman Poole wave drag parameter

P_c = cushion pressure, lb/ft^2

W = displacement, lbs

ρ_w = density of water, 1.99 $slugs/ft^3$

g = acceleration due to gravity, 32.2 $(ft\text{-}lbm)/(lbf\text{-}sec^2)$

L_c = cushion length, ft

F_L = Froude number, $V/(g L_c)^{\frac{1}{2}}$

V = ship velocity, ft/sec

Figure 5 shows the effect of L_c/B_c on cushion wavemaking drag. The large hump in the drag parameter curve which appears at $F_L \approx 0.60$ for low L_c/B_c craft essentially disappears for high L_c/B_c craft. As the Froude number increases to values greater than 1.0, however, the wave drag parameter for low L_c/B_c craft is less than for high L_c/B_c craft.

The hydrodynamic resistance of the sidehulls is divided into three components; sidehull wavemaking drag, sidehull frictional drag, and sidehull form drag. The sidehulls for most SES's tend to be very slender. For the 5000 ton SES being considered, the ratio of sidehull beam to cushion length is $7/450 \approx 1/64$ (The sidehull beam selected is in the same range as the beam proposed for the 3000 ton SES). Due to these high fineness ratios and the low draft when these vehicles are "on cushion", the sidehull wavemaking drag is almost negligible. The sidehull wavemaking drag coefficient can be calculated based on the wave resistance theory for thin bodies (Reference 3) as follows:

$$D_{SHW} = 2\ C_r\ \frac{8\rho_w\ g\ B_{SH}^2\ h^2}{\pi\ L_S} \tag{3.2}$$

where D_{SHW} = sidehull wavemaking drag, lbs

C_r = sidehull wavemaking drag coefficient which is plotted in Figure 6 as a function of the reciprocal of Froude number squared for a parabolic thin body and a "full" form. The "full" form is used for vessels with a transom stern and is used in this analysis.

B_{SH} = sidehull beam, ft.

h = $h_w' + \frac{1}{2}(h_b)$ = average draft, ft.

h_w' = $\frac{1}{2}$ average wave height h_w, ft.

h_w = average wave height, ft.

h_b = P_c/ρ_w $(lbs/ft^2)/(lbs/ft^3)$ = water depression due to cushion pressure, ft.

L_S = maximum wetted sidehull length, ft.

for sea state 3, h_w = 2.9 ft

h_w' = 1.45 ft

h_b = 5.07 ft

h = 3.99 ft

Sidehull frictional drag is a function of the viscosity of the water, hull roughness, and the wetted surface area of the sidehulls. The SES wetted sidehull geometry is shown in Figure 5a. The total wetted surface can be separated into two distinct areas. The first is the rectangular surface with sides h_w' and L_S. Since each sidehull has an inner and outer surface which is wetted, there are a total of four areas:

$$(SA)_1 = 4\, h_w'\, L_S \tag{3.3a}$$

The second surface area which must be considered is the triangular area which is wetted on the outside of both sidehulls:

$$(SA)_2 = 2\left(\frac{h_b}{2}\right)L_S = h_b\, L_S = L_S^2\,\frac{h_b}{L_S} \tag{3.3b}$$

From Figure 5a it can be seen that:

$$\frac{h_b}{L_S} = \frac{D_{WM}}{W} = \frac{D_{WM}}{P_c\, S_c} \tag{3.3c}$$

The total surface area is:

$$(SA)_{TOT} = (SA)_1 + (SA)_2 = 4\, h_w'\, L + \frac{D_{WM}\, L_S^2}{P_c S_c} \tag{3.3d}$$

The frictional drag for the smooth surface of SES sidehulls is the product of the drag coefficient C_f, water dynamic pressure $\frac{1}{2}(\rho_w\, V^2)$, and the wetted area:

$$D_{SHF} = \frac{C_f \rho_w\, V^2}{2}\quad 4\, h_w'\, L_S + \frac{D_{WM}\, L_S^2}{P_c S_c} \tag{3.3e}$$

where D_{SHF} = sidehull frictional drag, lbs.

C_f = friction factor for smooth surfaces, $\dfrac{0.075}{(\log_{10} R_e - 2)^2}$

R_e = Reynolds number, $V\, L_S/\nu_w$

V = ship speed, ft/sec

L_S = sidehull length submerged, ft.

ν_w = Kinematic viscosity of sea water, 1.28×10^{-5} ft^2/sec at 59°F

S_c = cushion area, ft^2

The sidehull form drag due to the shape of the sidehull and the re-

sulting pressure drag can be determined from:

$$D_{SFD} = 1.16 \frac{B_{SH}}{L_S} D_{SHF} \tag{3.4}$$

where D_{SFD} = sidehull form drag, lbs.

B_{SH} = beam of sidehull, ft.

L_S = sidehull length submerged, ft.

The aerodynamic drag of an SES is primarily the profile drag, which is a function of the frontal area, the speed, and the profile drag coefficient. The value of the profile drag coefficient varies with the streamlining of the craft. The profile drag coefficient for a well streamlined body is about 0.25, while the drag coefficient increases to 0.38 for poorly streamlined shapes (Reference 4). The aerodynamic drag of an SES can be written as:

$$D_A = C_{DO} \tfrac{1}{2}\rho_a V_r^2 S_F \tag{3.5}$$

where D_A = aerodynamic drag, lbs.

C_{DO} = aerodynamic drag coefficient

S_F = reference frontal area for C_{DO}, ft^2

ρ_a = mass density of air, 0.0024 $(lb\text{-}sec^2)/ft^4$

V_r = relative wind velocity impinging upon the frontal area, ft/sec.

V_r = $V + V_w$

V = ship speed, ft/sec.

V_w = headwind velocity, ft/sec (27.02 in SS-3)

For the SES in this analysis:

C_{DO1} = 0.35 - refers to the hull and the area below the 01 level (36 ft)

C_{DO2} = 0.25 - refers to the two decks of the superstructure (18 ft)

S_{F1} = frontal area of the hull which includes the overall beam and the height to the 01 level. $(69.23 + 7 + 7)(36) = 2996 \text{ ft}^2$

S_{F2} = superstructure frontal area - about $\frac{1}{4}$ of the total frontal area (749 ft^2)

Combining these terms in the equation for aerodynamic drag:

$$(D_A)_{HULL} = \frac{(0.35)(0.0024)(V + V_w)^2(4494)}{2} = 1.2583(V + 27.02)^2$$

$$(D_A)_{SUP} = \frac{(0.25)(0.0024)(V + V_w)^2(1123.6)}{2}$$

$$= 2.2471 \times 10^{-1}(V + 27.02)^2$$

$$(D_A)_{TOT} = (D_A)_{HULL} + (D_A)_{SUP} \qquad (3.6)$$

The ram drag or momentum drag is that force which arises from bringing a constant mass flow rate of air from a non zero velocity relative to the SES to a zero velocity relative to the vehicle. The ram drag may be written as:

$$D_R = 2\, D_C \frac{S_g}{S_c} \left(\frac{q_a}{\omega}\right)^{\frac{1}{2}} W \qquad (3.7)$$

where D_R = ram or momentum drag, lbs.

D_C = air leakage or discharge coefficient (assume $D_C = 0.65$ from Reference 4 for plenum craft)

$S_g = 2\, B_c\, h_g$ = cushion discharge area, ft^2

h_g = cushion gap, ft. (assume $h_g = 0.333$ ft)

S_c = cushion area, ft^2

$q_a \;= \frac{1}{2}\;{}_a V^2$ = air dynamic pressure, lb/ft^2

$W \;= P_c S_c$ = ship displacement supported by cushion air pressure, lbs.

$\omega \;= W/S_c$ = specific weight of SES, lb/ft^2.

The equation for ram drag can be rewritten and simplified as follows:

$$D_R = 2\,D_C \frac{S_g}{S_c} \left(\frac{\frac{1}{2}\rho_a V^2 S_c}{W}\right)^{\frac{1}{2}} W = 2\,D_C \frac{S_g}{S_c} \left(\frac{\frac{1}{2}\rho_a V^2 S_c (P_c S_c)^2}{P_c S_c}\right)^{\frac{1}{2}}$$

$$D_R = 2\,D_C \frac{S_g}{S_c} \left(\tfrac{1}{2}\rho_a V^2 S_c P_c S_c\right)^{\frac{1}{2}} = 2\,D_C \frac{S_g}{S_c} \left(\tfrac{1}{2}\rho_a V^2 P_c\right)^{\frac{1}{2}} S_c$$

$$D_R = 2\,D_C\,S_g\,V \left(\tfrac{1}{2}\rho_a P_c\right)^{\frac{1}{2}} \tag{3.8}$$

The bow and stern seal drag components make up a substantial part of the rough water resistance of an SES. The selection of gap clearance between the seals and the surface of the water involves a trade off between lift power and propulsive power. A small value of gap clearance means lift power is greatly reduced. However, as the SES operates in a seaway, the flexible seals encounter waves and drag through the water resulting in an increase in propulsive power required. Most of the methods for estimating the drag due to the seals are derived from empirical data resulting from model tests conducted by the British hovercraft industry. The seal drag for a fully skirted air cushion vehicle (ACV) can be written as:

$$D_S = \frac{C_{DS}\,S_c\,q_a\,W}{L_c\,\omega\,B_c} = \frac{C_{DS}\,q_a\,W}{\omega}$$

$$D_S = \frac{0.0012\,C_{DS}\,q_w\,W}{\omega} = \frac{0.0012\,C_{DS}\rho_w V^2 S_c}{2} \tag{3.9}$$

where D_S = seal drag for an ACV, lbs.

q_a = dynamic air pressure ($\frac{1}{2}\rho_a V^2$) lb/ft^2

q_w = dynamic water pressure ($\frac{1}{2}\rho_w V^2$) lb/ft^2

C_{DS} = seal drag coefficient

The seal drag coefficient C_{DS} is a function of the average wave height, the cushion length, and the gap clearance. The drag coefficient is the drag per unit cushion area S_c, and per unit air dynamic pressure q_a:

$$C_{DS} = 6.6\left(\frac{h_w - 2h_g}{L_c}\right)^{1.2} \tag{3.10}$$

where h_w = average wave height, ft (5.8 ft for SS-3)

h_g = gap clearance height, ft (0.333 ft)

when $h_w - 2h_g \quad 0 \qquad C_{DS} = 0$

$h_w - 2h_g \quad 0 \qquad C_{DS} = 6.6 \left(\frac{h_w - 2h_g}{L_c}\right)^{1.2}$

Since the equation for seal drag was derived from empirical data for full peripheral air cushion vehicle (ACV) skirts, and since the SES uses only a forward and after seal, a geometric factor (G) is used to relate the ACV skirt drag to SES seal drag. The geometry factor (G) depends on the L_c/B_c of the SES and is given in Reference 5 as $1/(1 + L_c/B_c)$. The complete expression for SES seal drag can then be written as:

$$D_S = \frac{0.0012\ C_{DS}\rho_w V^2 S_c G}{2} \tag{3.11}$$

The total craft smooth water drag is the sum of cushion wavemaking, sidehull wavemaking, sidehull frictional, sidehull form, aerodynamic, ram or momentum, and seal drag (Appendage drag will not be calculated separately, but will be included in the rough water multiplication factor) :

$$(D_{TOT})_{SW} = D_{WM} + D_{SHW} + D_{SHF} + D_{SFD} + D_A + D_R + D_S \qquad (3.12)$$

where $(D_{TOT})_{SW}$ = total craft smooth water drag, lbs.

For operation in the open ocean, the smooth water drag is increased by a rough water drag factor. It is assumed that this factor is 10% for sea state 3 and includes the drag due to the two rudders forward of the propulsors. Then the total craft drag is:

$$(D_{TOT})_{RW} = 1.10\ (D_{TOT})_{SW} \qquad (3.13)$$

where $(D_{TOT})_{RW}$ = total craft rough water drag, lbs.

This value will be used in computing the propulsive horsepower requirements. The graph of Figure 7 gives a summary of the SES drag over several values of Froude number.

4. PROPELLER SELECTION

In the two speed regimes of interest, 50 knots in a calm sea and 45 knots in SS-3, the highest calculated overall propulsive efficiency system incorporates a partially submerged supercavitating propeller (Reference 6). The partially submerged supercavitating propeller (PSSCP) may be mounted either horizontally or on an inclined shaft as shown in Figure 8. The relatively thin sidehull beam of the SES being considered (7 feet) cannot accomodate the engines, transmission, and propulsor. This means propulsive power must be transmitted from the prime mover to the remotely mounted propulsor through a system of right angle gears and shafts similar to the system used in the SES-100B.

An alternative arrangement is to use an inclined shaft with raked propeller blades such that the blades are perpendicular to the surface of the water when they are submerged. The inclined shaft offers certain machinery arrangement advantages over the horizontal shaft. The horizontal shaft shown in Figure 8 requires three right angle bevel gears. This number could be reduced to two by using the inclined shaft arrangement. The disadvantages of the inclined shaft arrangement include rigidity problems resulting from the long section of shafting from the wet deck to the propulsor. This results in an increase in the unsteady propeller loading forces. Another factor which must be considered in an inclined shaft arrangement is the engine manufacturer's limits on the trim angle at which the gas turbines can be operated. Typical values for the permanent trim of modern gas turbines are 10-15 degrees. An engine inclination angle which exceeds this range would require engine modifications, especially

in the lubrication and bearing systems. In spite of these technical problems, the inclined shaft arrangement was selected for this study on the basis of the potential machinery arrangement advantages.

Although there is a limited amount of data available on the effects of shaft inclination and blade rake angles on supercavitating propellers, References 6 and 7 discuss two propellers tested at NSRDC at various angles of shaft inclination and rake angle. While the detailed effects of shaft inclination and rake angle are strongly dependent upon the entire propeller-power plant-transmission interaction, some general conclusions about these systems are possible (Reference 7) :

1. Inclining the shaft slightly $\lesssim$ 20 degrees tends to slightly increase propeller efficiency.
2. At inclination angles greater than 20 degrees, the efficiency, torque, and propeller thrust decrease.
3. The higher the rake angle for a given shaft angle the lower the efficiency, thrust, and torque.
4. The higher the pitch angle, the greater are the effects of shaft inclination and rake angle.

For the SES under consideration there are several other criteria. First, the propeller must be controllable-reversible pitch (CRP). This requirement dictates the need for an expanded area ratio $\lesssim$ 0.75. The hub/diameter ratio required for the CRPP mechanism of an eight bladed propeller is 0.45. The two NSRDC propellers discussed in Reference 7 satisfy these criteria.

The methodology used to select a propeller from the design data pro-

vided in Reference 7 is outlined below :

1. Calculate the required horsepower.

$$(HP)_{BR} = \frac{(D_{TOT})_{RW} V}{(550)(NPC) \gamma_T} \tag{4.1}$$

where $(HP)_{BR}$ = horsepower required

$(D_{TOT})_{RW}$ = total rough water drag, lb.

V = ship speed, ft/sec

NPC = $\gamma_o \cdot \gamma_H \cdot \gamma_R \cdot \gamma_M$, net propulsive coefficient

γ_o = propeller open water efficiency

γ_H $= \frac{1 - t}{1 - w} = \frac{1 - \text{thrust deduction fraction}}{1 - \text{wake fraction}}$

= hull efficiency

γ_R = relative rotative efficiency

γ_M $= 1 - \frac{D_M}{(D_{TOT})_{RW}}$, mounting efficiency

D_M = propulsor mounting drag

γ_T = transmission efficiency (assume 0.97)

For this initial stage of the design, it was assumed that $\gamma_H = \gamma_M = \gamma_R = 1.00$. Then NPC = γ_o. Equation (4.1) combined with equations (3.12) and (3.13) results in :

$$(HP)_{BR} = \frac{3.667 \times 10^4}{NPC} \quad \text{- for 45 knots in SS-3} \tag{4.2a}$$

$$(HP)_{BR} = \frac{4.116 \times 10^4}{NPC} \quad \text{- for 50 knots in SS-0} \tag{4.2b}$$

It can be seen that the requirement to make 50 knots in SS-0 sets the upper limit on propulsive power required, therefore this larger value will be

used to select the propeller. The propeller selected must then satisfy the following values :

a. Total thrust = 260,000 lb

(Thrust/propulsor = 130,000 lb)

b. Total required horsepower $= \dfrac{4.116 \times 10^4}{\gamma_o}$

(Required horsepower/propulsor $= \dfrac{4.116 \times 10^4}{2\,\gamma_o}$)

2. Select the percent submergence, shaft inclination, and rake angle to be investigated.

3. Select a value of NPC(γ_o).

4. Determine the corresponding values of J, K_T, and K_Q.

$$J_P = \frac{V_a}{n\,D_P}$$

where J_P = advance coefficient

V_a = speed of advance, ft/sec

n = revolutions per second

D_P = propeller diameter, ft.

$$K_Q = \frac{Q_P}{\rho_w\, n^2\, D^5}$$

where K_Q = torque coefficient

Q_P = propeller torque, ft-lbs.

ρ_w = density of water, 1.99 slugs/ft^3

$$K_T = \frac{T}{\rho_w\, n^2\, D^4}$$

where K_T = thrust coefficient

T = thrust, lb.

$$(HP)_{BP} = \frac{2\pi\ n\ Q_P}{550\ \gamma_R\ \gamma_T}$$

where $(HP)_{BP}$ = propeller horsepower

5. Select values of n and D_P and try and match the given thrust, torque, and horsepower required.

6. Iterate until all values are simultaneously satisfied.

An important consideration in this selection procedure is the propeller RPM. Epicyclic reduction gears will be used in the design because of the light weight and compactness of these gears. The maximum recommended reduction ratio for simple planetary gears is 13.4 : 1.0 (Reference 8). For a 3600 RPM input shaft this means an RPM of greater than or equal to 269. Therefore, the RPM of any propeller selected should be $\geqslant$ 269 in order to stay within the state-of-the-art for planetary gears.

A summary of feasible propellers is provided in Table 3. The propeller which best satisfies the criteria is NSRDC 4281 with 30% submergence, 20° shaft inclination and a 20° rake angle. The open water propeller efficiency of 0.70 gives a value of $(HP)_{BP}$ = 30,990. Comparing this to equation (4.2b) gives a value for NPC of 0.66, and $\gamma_M\ \gamma_H\ \gamma_R = 0.94$. These values are with acceptable limits for state-of-the-art in the 1980's. The propeller selected has an RPM of 375 which is greater than the specified minimum value of 269.

5. LIFT SYSTEM DESIGN

The primary function of the lift system is to maintain a cushion of air beneath the SES to support it. The lift system must also have the flow capacity to compensate for the leakage of air through the gap between the flexible bow and stern seals and the water's surface. As the vehicle moves forward, the leakage rate tends to increase with increasing speed. Also, as the sea state increases, the waves tend to pump the cushion air out more rapidly. The major components of the fan system are :

1. The inlet - which brings air into the fans
2. The fans - which add velocity and pressure to the air
3. The diffuser - which converts the air velocity to pressure
4. The ducting - which provides a path for the air from the fans to the cushion plenum
5. The control system - which allows operation of the fans to ensure peak efficiency and adequate ride quality

There are three primary types of fan considered for installation on an SES : axial, centrifugal, and mixed-flow fans. The general arrangement geometry of each fan type is shown in Figure 9. Axial fans can be operated at efficiency levels of 80-85%. The rotational speed of axial fans is higher than centrifugal fans operating at the same pressure and flow rate. These high rotational speeds have the advantage that gear reduction ratios from the engines to the fans are minimized and ducts can be kept small. However, these high rotational speeds mean high noise levels. Axial fans lend themselves to controllable pitch blade systems which improve the off-design performance of the fan and can be integrated into the ride control

system.

Centrifugal fans generally operate at lower speeds than axial fans with the same pressure and flow requirements. These fans tend to be larger and heavier than their axial counterparts. Controllable pitch blades have not been developed for centrifugal fans, however, variable pitch inlet vanes have been proposed for improving the off-design performance of these fans. Centrifugal fans for SES application have airfoil shaped blades with three types of curvature : forwardly curved, radial, and backward curved blades as illustrated in Figure 10. Airscrew-Weyroc has developed the HEBA series of centrifugal fans. The HEBA (high efficiency backward foil) fans were developed in two basic forms : HEBA-A, a narrow-width fan suitable for high pressure, and HEBA-B, a wide-width fan suitable for lower pressures. The HEBA-B series fans are especially applicable to SES installations due to the pressure flow characteristics and have demonstrated improved efficiency over conventional flat sheet metal blades. The SES-100B, JEFF (A) and JEFF (B) craft all use variants of the HEBA-B fans. Centrifugal fan systems lend themselves to mounting several fans on a single shaft. The axis of the shaft could run parallel to the sidehulls as shown in Figure 11. This arrangement greatly reduces the transmission complexities of the fan system. This approach has been proposed for the 3000 ton LSES and was arbitrarily selected as the fan system to install in the 5000 ton SES being analyzed in this study.

Mixed flow fans have performance characteristics which fall between those of the centrifugal and axial fan types. While efficiencies similar to axial and centrifugal fans are theoretically possible, these efficiencies

are difficult to obtain in actual practice due to fabrication problems. Another problem in utilizing a mixed flow fan system is the efficient distribution of air from the fan exit. This is because the flow exiting from the mixed flow fan has axial, tangential, and radial velocity components.

A common parameter used to describe all three fan types is the non-dimensional term, N_s, specific speed :

$$N_s = \frac{N\, Q^{\frac{1}{2}}}{H^{3/4}} = \frac{N\, Q^{\frac{1}{2}}}{(P_c/\rho_a)^{3/4}} \tag{5.1}$$

where N_s = specific speed

N = RPM

Q = fan air volume flow rate, ft^3/sec

H = fan discharge head rise, ft.

ρ_a = air density, 0.073 lb/ft^3

P_c = cushion pressure, lb/ft^2

Impeller geometry and dynamic similarity for scaling can be based upon a common N_s value. Centrifugal fans are usually designed for $N_s < 150$, however, HEBA-B fans have been designed for values of N_s as high as 300. Mixed-flow fans are essentially applied at moderate values of N_s ($150 < N_s < 400$). Axial fans are normally applied at high values of N_s ($N_s > 400$). Actual fan designs show a large degree of latitude in selecting a value of N_s for a given fan type. High N_s values are desirable because as N_s increases, impeller diameter decreases, which results in a decrease in the size and weight of the fan. As design values of N_s exceed 300, centrifugal fans approach noise and stress limits due to the high tip speeds. Axial fans can operate at very high N_s ($N_s > 400$) and thereby reduce the size

and weight of the fans at the aximum allowable tip speed. However, axial impeller dynamic efficiencies decrease significantly as N_s exceeds 800.

Fan size can also be non-dimensionalized by the specific diameter, D_{SF}, which is defined as :

$$D_{SF} = \frac{D\ (P_c/\rho_a)^{\frac{1}{4}}}{Q^{\frac{1}{2}}} \tag{5.2}$$

where D_{SF} = specific diameter

D = fan impeller diameter, ft.

Optimum fan designs (maximum total efficiency) of axial, centrifugal, and mixed-flow fans can be represented by single curves of specific diameters (D_{SF}) and specific speed (N_s). Cordier (Reference 9) compiled specific diameter and specific speed data of high efficiency fans and showed that the data exhibited very little scatter. Balje (Reference 10) and others have derived a theoretical prediction for the same information Cordier derived empirically. Figure 12 (Reference 11) compares Cordier's empirical results and Balje's theoretical results. Figure 12 also includes data points for recent SES and ACV fan designs and shows that recent fans for these vehicles have been designed to maximize efficiency. Also available from Reference 11, is Figure 13 which presents the theoretical maximum obtainable efficiencies and average maximum values of fan efficiencies obtained empirically. In actual SES installations maximum obtainable fan efficiencies have been 80-85%. From these curves it is possible to describe fan size in terms of the characteristic specific speed by (Reference 11) :

$$D_{SF} = 1.20 + 1.90\ (\frac{N_s}{60})^{-1.24} \tag{5.3}$$

Equation (5.3) is applicable over the range $20 \leq N_s \leq 250$ and covers axial, centrifugal, and mixed-flow fans. This sizing equation also insures the maximum efficiency of the sizes used. Fans designed from equation (5.3) tend to be large in diameter in order to maximize efficiency. Table 4 (Reference 11) illustrates the trade-off of size versus efficiency. A reduction of 10% in maximum efficiency results in a 20% reduction in fan diameter and a 40% reduction in fan weight.

Another criteria which must be included in any fan design is the tip speed, U_t :

$$U_t = \frac{\pi N D}{60} \tag{5.4}$$

where U_t = tip speed, ft/sec

N = rotative speed, RPM

D = impeller diameter, ft.

$$\text{Or } U_t = (gH/\psi)^{\frac{1}{2}} \tag{5.5}$$

where g = gravity constant, 32.2 ft/sec^2

$H = (P_c/\rho_a) = (P_c/0.073)$ = fan head, ft

$$\psi = \frac{P_c\, g}{\rho_a} = \text{head coefficient}$$

ρ_a = air density at sea level = 0.073 lb/ft^3 (80°F is used as the ambient temperature for all fan calculations)

Tip speed limits are indicators of structural or noise limits. The recommended limits of tip speed for lift fans is summarized in Table 5 (Reference 12).

The selection of fan type for a given set of lift requirements involves a consideration of impeller design limits and a trade-off of weight

and size versus efficiency for the particular installation under consideration.

The method used to design the fans in this analysis is outlined below:

1. Calculate the flow requirements (Q) for the SES. The cushion air flow rate must be calculated for both the static lift and the wave pumping case. Whichever case results in the larger value of air flow requirements is used to determine the cushion powering requirements.

a. Static Lift Case

$$Q_{SL} = S_g D_C \left(\frac{2 P_c}{\rho_a} \right)^{\frac{1}{2}} \tag{5.6a}$$

where Q_{SL} = volume flow rate of air in state lift, ft^3/sec

S_g = $2B_c h_g$ = cushion discharge area, ft^2

B_c = cushion beam, ft

h_g = cushion gap, ft (assume h_g = 0.3333 ft)

D_C = air leakage discharge coefficient (D_C = 0.65 for plenum craft, Reference 4)

ρ_a = mass density of air = 0.0024 (lb-sec^2)/ft^4

P_c = cushion pressure, lb/ft^2

For the case being studied :

$$Q_{SL} = 2(69.23)(0.3333)(0.65)\left(\frac{(2)(325)}{0.0024}\right)^{\frac{1}{2}} = 15{,}611 \ \frac{\text{ft}^3}{\text{sec}} \tag{5.6b}$$

b. Wave Pumping Case

$$Q_{WP} = B_c h_w V \tag{5.7a}$$

where Q_{WP} = volume flow rate of air in a seaway due to the wave pumping action of the sea, ft^3/sec

B_c = cushion beam, ft

h_w = wave height, ft (2.9 ft in SS-3)

V = ship speed, ft/sec

For the case being studied :

$$Q_{WP} = (69.23)(2.9)(1.689)(V \text{ knots})$$

$$Q_{WP} = 339.10 \ (V \text{ knots}) \qquad (5.7b)$$

Combining (5.6b) and (5.7b) and solving for speed shows that at speeds below 46 knots, the static flow requirements dominate. Since the speed regime of interest is 45 knots, the static flow requirements will be used in all future calculations.

2. Select the number of fans to be installed in the SES. For the 5000 ton SES, eight fans were selected arbitrarily. (The proposed 3000 ton LSES uses six fans.) The volume air flow rate per fan is :

$$\frac{Q_{SL}}{8} = \frac{15,611}{8} = 1951 \text{ ft}^3/\text{sec}$$

3. Select fan tip speed. For the 5000 ton SES 600 ft/sec was selected. This value is slightly higher than the stress related conservative speed limit for centrifugal fans but less than the noise related limit of 800 ft/sec.

4. Determine specific speed N_s and specific diameter from Figure 12. In the case being studied, HEBA-B centrifugal fans have already been selected because of their high efficiency and the arrangement advantages.

For HEBA-B fans : $\frac{N_s}{60} = 1.70$

$$D_{SF} = 2.00$$

The fan efficiency for these fans from Figure 13 is 82%.

5. Calculate fan RPM, N.

$$N = \frac{N_s(P_c/\rho_a)^{3/4}}{Q_f^{\frac{1}{2}}} \tag{5.8a}$$

where N = fan speed, RPM

N_s = specific speed

P_c = cushion pressure, lb/ft^2

ρ_a = mass density of air, 0.073 lbs/ft^3

Q_f = $(Q_{SL})/n_f$ = volume air flow rate per fan, ft^3/sec

n_f = number of fans

For the case being studied :

$$N = \frac{100\ (325/0.073)^{3/4}}{(1951)^{\frac{1}{2}}} = 1233.9 \text{ RPM} \tag{5.8b}$$

6. Calculate fan diameter, D.

$$D = \frac{D_{SF}\ Q_f^{\frac{1}{2}}}{(P_c/\rho_a)^{\frac{1}{4}}} \tag{5.9a}$$

where D = impeller diameter, ft.

D_{SF} = specific fan diameter

Q_f = volume air flow rate per fan, ft^3/sec

P_c = cushion pressure, lbs/ft^2

ρ_a = air density, 0.073 lbs/ft^3

For the case being studied :

$$D = \frac{(2.00)(1951)^{\frac{1}{2}}}{(325/0.073)^{\frac{1}{4}}} = 10.8 \text{ ft.} \tag{5.9b}$$

This value is large compared to existing and proposed SES fan designs. In order to reduce the size of the fans Table 4 was used to trade-off size

against efficiency. By using $\gamma_f / \gamma_{max} = 0.95$, the new fan diameter was calculated as :

$$\gamma_{max} = 0.82$$

$$\gamma_f = 0.78$$

$$D_{SF} = 1.07 + 1.33 \left(\frac{N_s}{60} \right)^{-1.85} = 1.57 \qquad (5.10a)$$

Substituting into equation (5.9b),

$$D = 8.56 \text{ ft}$$

Thus a 40% reduction in total efficiency resulted in a 20% reduction in size. This impeller diameter value is more in line with proposed SES fan designs and will be used in all future calculations.

7. Determine other fan dimensions. These dimensional proportions and size relationships were developed in Reference 12 and are illustrated in Figure 14.

$$L = 0.80 \, D \qquad (5.11a)$$

where L = overall fan length, ft

D = impeller diameter, ft

$$A = 1.87 \, D \qquad (5.11b)$$

where A = overall fan diameter, ft

For the case being studied :

$$L = 6.86 \text{ ft}$$

$$A = 16.01 \text{ ft}$$

8. Estimate the weight of each fan.

$$W_f = 4 \, D^2 \quad \text{(Reference 12)} \qquad (5.12)$$

where W_f = weight of a single entry centrifugal fan, lbs.

D = impeller diameter, ft.

For the case being studied :

$$W_f = (4)(8.56)^3 = 2509 \text{ lbs.}$$

9. Calculate the shaft horsepower required for the lift system.

$$(HP)_C = \frac{(P_c + \Delta P_c)\ Q}{550\ \gamma_f\ \gamma_d\ \gamma_{TF}} \tag{5.13}$$

where $(HP)_C$ = required lift system horsepower

P_c = cushion pressure, lb/ft^2

ΔP_c = variation in cushion pressure, $0.05\ P_c\ lb/ft^2$

Q = total volume air flow rate, ft^3/sec

γ_f = fan efficiency ($\gamma_f = 0.78$ from step 6 above)

γ_d = duct efficiency (assume $\gamma_d = 0.90$)

γ_{TF} = fan transmission efficiency (assume $\gamma_{TF} = 0.98$)

For the case being studied :

$$(HP)_C = \frac{(1.05)(325)(15{,}611)}{(550)(0.78)(0.90)(0.98)} = 14{,}079 \text{ H.P.}$$

A summary of the lift fan characteristics for the 5000 ton SES is provided in Table 6.

6. ENGINE SELECTION

The horsepower requirements have now been determined for both the propulsion system and lift system. There are two plant arrangements, split and combined, and engines must be selected for both systems which most nearly match the power requirements.

The engines for the combined plant will be selected first. The engines for the combined plant must provide power for both the lift and propulsion systems. The lift power requirement is for 14,079 H.P. and the propulsion power requirement is for 62,000 H.P. (31,000 H.P. per shaft). The source for engine data is Table 7 (Reference 13). Since the vessel being analyzed should make use of the most advanced technology available, only those engines which are in the category of "second generation" engines. Figure 15 (Reference 14) provides design lanes for identifying first and second generation gas turbines. Second generation engines incorporate turbine blade and vane cooling, generally operate at higher pressure ratios. These more modern gas turbines also have higher power-to-weight ratios and have lower fuel consumption rates than first generation turbines with the same power levels. Using these criteria, the engine selected for the combined plant is the Pratt and Whitney FT9 which is rated at 40,000 continuous H.P. Two engines would provide 80,000 H.P. The total power requirement of the 5000 ton SES is 76,079 H.P. (62,000 + 14,079). If each of two FT9 gas turbines is operated at 38,040 H.P., the SFC will be degraded slightly. The following equation (Reference 14) was used to estimate the new SFC at the 38,040 H.P. level :

$$f' = (0.491\, n' + 0.316)\, t' + 0.196\, n' - 0.004 \qquad (6.1)$$

where f' = (fuel rate)/(design fuel rate)

n' = RPM/(design RPM)

t' = torque / design torque

Fuel rate = H.P. / SFC

Torque = (H.P.)(550) / (2π)(RPM/60)

For the case being analyzed :

$n' = 1.000$

$t' = 0.951$

$f' = 0.959$

The new fuel rate is calculated from :

$$\text{New fuel rate} = (f' \times \text{SFC})(\text{H.P.}) \qquad (6.2)$$

For the case being analyzed :

$$\text{New fuel rate} = (0.959)(0.37)(40{,}000) = 14{,}199.8\ \frac{\text{lbs}}{\text{HR}}$$

The new SFC is calculated from :

$$\text{New SFC} = (\text{new fuel rate}) / (\text{new H.P.}) \qquad (6.3)$$

For the case being analyzed :

$$\text{New SFC} = (14{,}199.8\ \text{H.P.}) / (38{,}040) = 0.3733\ \frac{\text{lbs}}{\text{HP-HR}}$$

In summary, the combined plant will consist of two FT9 engines operating at 38,040 H.P. each, at 3600 RPM, and an SFC of 0.3733 $\frac{\text{lbs}}{\text{HP-HR}}$ at the speed of 50 knots in a calm sea.

The split plant will utilize separate engines for the lift and propulsive functions. The propulsive horsepower requirement is for 31,000 horsepower per shaft. Examining Table 7, there is no second generation gas turbine which exactly meets the requirement. (The FT4C-2, while nearly

meeting the horsepower requirements does not fall within the design lanes for second generation engines in Figure 15.) The second generation gas turbine which most closely meets the propulsive power requirements is the General Electric LM 3500 rated at 34,950 horsepower. Operating this engine at 31,000 horsepower will degrade the SFC slightly. Using equation (6.1) :

$$n' = 1.000$$

$$t' = 0.8870$$

$$f' = 0.9078$$

Similarly, applying equations (6.2) and (6.3) :

New fuel rate = 12,056 lbs/HR

New SFC = 0.3889 $\frac{\text{lb}}{\text{HP-HR}}$

In the split plant, the fan system requires 14,079 horsepower. For redundancy, a minimum of two turbines is necessary. If two turbines are used, each turbine must provide 7024 horsepower. The second generation gas turbine which most nearly meets these requirements is the Garrett - AiResearch GPTF990 rated at 5900 continuous horsepower. If three engines are used for lift, the General Electric LM500 rated at 4500 horsepower each would meet the requirements.

Looking first at the two turbine case, the two GPTF990 turbines at 5900 horsepower each would not satisfy the 7040 horsepower required. However, for the purposes of this analysis and in order to compare the two turbine fan system with the three turbine fan system, it was assumed that the GPTF990 engines could be upgraded to provide 7040 horsepower each. It was further assumed that the GPTF990 operated at 7040 horsepower would operate at the same SFC and RPM. The weight of the upgraded GPTF990 engines

was calculated using the specific weight listed in Table 7 :

$$\text{New turbine weight} = (0.87 \frac{\text{lb}}{\text{H.P.}})(7040 \text{ H.P.}) = 6125 \text{ lbs.}$$

In the three turbine case, each turbine must contribute $\frac{14{,}079}{3}$ = 4693 horsepower. It was assumed that the LM500 engines could be upgraded from 4500 to 4693 horsepower. The new SFC and fuel rate are determined by applying equations (6.1), (6.2), and (6.3) :

$$n' = 1$$

$$t' = 0.964$$

$$f' = 0.970$$

$$\text{New fuel rate} = 2008 \text{ lbs/HR}$$

$$\text{New SFC} = 0.430 \frac{\text{lbs}}{\text{HP-HR}}$$

A summary of the characteristics for all engines selected for both the propulsion and lift system is provided in Table 8.

7. TRANSMISSION DESIGN

In previous sections the propulsors, lift fans, and engines were selected for the 5000 ton SES. The next stage of the analysis is a critical one - the size and weight estimation of the transmission system.

Before proceeding, it is necessary to establish the degree of redundancy and cross-connection capability to be designed into the transmission system. For the purposes of this analysis it will be assumed that the vessel will not have the capability to cross connect the propulsors. In other words, the SES will not have the capability of driving the port/starboard propeller with the starboard/port engine. The consequence of this assumption is that the SES must be able to hold a course with a single propeller. In the fan system, the split plant concept automatically incorporates some degree of redundancy in the lift fan system since the lift system is independent of the propulsive system. If a propulsor engine fails, the independent lift system will continue to function. In order to provide comparable redundancy in the combined plant, there must be a cross-connection capability to allow all lift fans to be operated from either of the two FT9 gas turbines. A schematic layout of the proposed transmission systems is illustrated in Figure 16.

Another important consideration is the location of the propulsive reduction gear and lift system reduction gear. The lightest weight transmission system results from placing the reduction gears as close as possible to the driven equipment and using very high rotational speeds for the shafting between the engines and the reduction gears. This approach is feasible for the fan system reduction gears. The fan reduction gear can

be located close to the fans. However, because of the narrow sidehulls in the SES, it will not be possible to install the propulsive reduction gear as close to the propulsor as desired for weight reduction purposes. Instead, the propulsion reduction gear for both the split and combined plants will be located on the wet deck, at the same level as the propulsion engines. Both the combined and the split plant will utilize the same shafting and reduction gear for the propulsor.

All gear and shafting parameters used in this analysis were derived from the gear and shafting size and weight estimating formulas summarized in Reference 8. A summary of these size and weight estimating equations is provided in Table 9 (Reference 8).

The starting point in the selection of gear and shaft parameters was the sizing of the planetary reduction gears used for the final gear reduction at the propulsive system and lift system. Earlier, in the selection of the propellers and fans, it was determined that the propeller RPM was 375 and the fan RPM was 1227. The overall reduction ratio M_g for each component is :

$$M_g = \frac{\text{RPM input}}{\text{RPM output}} = \frac{3600}{375} = 9.6 \quad \text{(propulsor)} \qquad (7.1a)$$

$$M_g = \frac{\text{RPM input}}{\text{RPM output}} = \frac{3600}{1227} = 2.93 \quad \text{(fans)} \qquad (7.1b)$$

From Table 10 (Reference 8), the number of planets (b) which result in the lightest weight reduction gear for the propulsor is three and for the fan system is eight. The nomenclature and terminology for these gears is summarized in Figure 17. The formula for the planetary gear pitch diameter d_s is :

$$d_s^2 F = \frac{12 \times 10^4 \, P(M_{gl} + 1)}{b \, K \, n_s \, M_{gl}} \tag{7.2}$$

where d_s = sun gear pitch diameter, in.

F = $2d_s$ = gear face width, in.

P = power, H.P.

K = $\dfrac{\overline{W}(M_g + 1)}{F \, d_p \, M_g}$ = gear surface durability factor, K-factor, lb/in²

b = number of planets (number of torque paths)

$\overline{W}$ = tangential gear load, lbs.

M_{gl} = d_p/d_s = $(M_g/2) - 1$ = first reduction ratio

n_s = speed of rotation, sun gear, RPM

d_p = planet pitch diameter, in.

Equation (7.2) can be reduced to :

$$d_s = 40 \left(\frac{P(M_g)}{b \, K \, n_s (M_g - 2)}\right)^{1/3} \tag{7.3}$$

The ring gear diameter (d_r) is calculated from :

$$d_r = d_s \, (M_g - 1) \tag{7.4}$$

The overall gear diameter (d), including the casing and bearings is :

$$d = 1.5 \, d_r \tag{7.5}$$

The gearbox length (l) is :

$$l = 3 \, F \tag{7.6}$$

Looking at equation (7.3), the K-factor can be seen to be a critical parameter. The K-factor reflects the surface durability of the gear. Using surface-hardened materials and higher accuracy in marine gear manufacture K-factors in the range 400-800 should be possible. A K-factor of 500 was

used throughout this study. The largest developmental planetary gear in the U.S. used a K-factor of nearly 1000 and transmitted 50,000 H.P. with an input speed of 3600 RPM. Using higher values for the K-factor reduces the size and hence the weight of planetary gears. The size and weight reduction effects of increasing the K-factor are important considerations for an SES. First, the SES is weight critical, less weight means less lift power is required. Secondly, the narrow sidehulls of an SES pose a serious arrangement problem for the installation of reduction gears. For the 5000 ton SES, the seven foot beam of the sidehulls is too narrow to accomodate the planetary reduction gear and the ancillary equipment and access space required. However, the effect of increasing the K-factor and reducing the size of the reduction gear could be very critical in other applications.

The empirical weight estimating relationships for planetary gears is based upon the K-factor and Q-factor :

$$W_P = 0.95 \times 10^4 \, (Q_D/K) \tag{7.7}$$

where W_P = weight of planetary reduction gears, lbs.

$$Q_D = \frac{P\,(M_g + 1)^3}{n_p\, M_g} = \text{Q-factor, H.P./RPM}$$

n_p = RPM of input pinion

The next transmission components to be sized will be the dual mesh spiral-bevel reduction gears used throughout the plant. These gears are used to transmit power and rotation between angularly displaced shafts. In most cases the angular displacement is 90°. Spiral-bevel gears were selected because they have smoother and quieter action than straight bevels. Dual mesh gears split the power and thus reduce the stress loading on each

gear, thus reducing the gear size required. Figure 18 illustrates the difference between dual mesh and single mesh gears.

Bevel gear size is affected by three factors; surface durability, tooth bending stress, and scoring. Scoring can be prevented with proper lubrication. The formula for the size of bevel gears limited by surface durability is :

$$d_p = \left(\frac{1600}{(1 + M_g^2)^{\frac{1}{2}}} \right)^{1/3} \left(\frac{P(M_g + 1)}{n_p \, b \, M_g} \right)^{0.32}$$

$$d_p = \left(\frac{1600}{(1 + M_g^2)^{\frac{1}{2}}} \right)^{1/3} R^{0.32} \quad , \quad R \leq 1.5 \qquad (7.8)$$

where d_p = pitch diameter of pinion gear, in.

b = number of torque paths ($b = 2$ for dual mesh)

$R = \left(\frac{P(M_g + 1)}{n_p \, b \, M_g} \right)$ = size factor

In the strength limited region :

$$d_p = \left(\frac{1400}{(1 + M_g^2)^{\frac{1}{2}}} \right)^{1/3} \left(\frac{P(M_g + 1)}{n_p \, b \, M_g} \right)^{0.42}$$

$$d_p = \left(\frac{1400}{(1 + M_g^2)^{\frac{1}{2}}} \right)^{1/3} R^{0.42} \quad , \quad R > 1.5 \qquad (7.9)$$

The gearbox dimensions can now be estimated from :

$$d_g = M_g \, d_p \qquad (7.10)$$

where d_g = gear pitch diameter, in.

d_p = pinion diameter, in.

$$d = 1.5 \, d_g \qquad (7.11)$$

where d = maximum gear diameter, in.

$$H_g = 2\, d_p \tag{7.12}$$

where H_g = gearbox height, in.

$$w_g = 1.5\, d_g \tag{7.13}$$

where w_g = gearbox width, in.

$$l = 2d_p \tag{7.14}$$

where l = gearbox length, in.

The dimensions of the second gear will be the same as the first for a 1 : 1 reduction ratio. All bevel gears in this analysis have a 1 : 1 reduction ratio. Bevel gear manufacturing facilities set a limit on the size of bevel gears available. Low ratio ($M_g < 3$) bevel gears are limited to a pitch diameter of 24 inches. For large reduction ratios ($M_g > 3$), the limit of the gear diameter is 33 inches. All high speed and heavily loaded gears must be finish ground after hardening. It is currently possible to grind bevel gears with a pitch diameter of 35 inches. It would be very expensive, however, gears as large as 35 inches could be made entirely by grinding. The weight of the bevel gears can be estimated from the equation :

$$W_B = 220\, Q_D^{0.8} \tag{7.15}$$

where W_B = weight of bevel gear, lbs.

$$Q_D = \frac{P(M_g + 1)^3}{n_p\, b\, M_g} = \text{size factor}$$

The weight of the transmission shafts which connect the prime mover, propulsor, and fans can make up a large percentage of the total transmission system weight. The parametric analysis for determining the size and weight of transmission shafts is based upon several simplifying assumptions. First, the shaft is assumed to be free floating and not subject to any significant axial, bending, or cyclic forces. In order to accomplish this

the shaft is separated into sections supported by bearings and connected by flexible couplings which absorb limited axial and angular changes. The propeller thrust is absorbed by the propeller thrust bearing. Using these assumptions, the size of the shaft is dependent only on its torsional strength. The spacing of the bearings and couplings, the critical length, will be determined from the shaft's critical speed. The coupling weight will be assumed to be a function of the torque. The bearings and miscellaneous weight are assumed to be a function of the shaft diameter.

The first step in the sizing of the transmission shafting is to calculate the shaft torque :

$$T' = \frac{P(550)(12)}{2\pi\,(RPM/60)} \tag{7.16}$$

where T' = shaft torque, lbs.-in

P = power, H.P.

The lightest-weight shafts are hollow torque tubes. The ratio of the outer diameter to the inner diameter is determined from Figure 19 (Reference 8), which plots the diameter ratio (f) for the minimum weight shaft as a function of torque. Also shown in Figure 18 is the torsional buckling limit for thin torque tubes. The buckling criteria for a long, slender torque tube is :

$$S_{CR} = \frac{0.22}{(1-\mu)^{3/4}}\; E\left(\frac{D_o - D_i}{D_o}\right)^{3/2} \tag{7.17}$$

where S_{CR} = critical shear stress, lb/in^2 (assume S_{CR} = 20,000 lb/in^2)

E = modulus of elasticity ($E = 30 \times 10^6$ lb/in^2 for steel)

μ = Poisson's ratio

Solving equation (7.17) for the diameter ratio (f) :

$$\xi = 1 - 2.74 \left(\frac{S_{CR}}{E} \right)^{2/3} (1 - \mu)^{\frac{1}{2}} \tag{7.18}$$

where $\xi = \dfrac{D_i}{D_o} = \dfrac{\text{(inner diameter)}}{\text{(outer diameter)}}$

The next step is to calculate the outer diameter of the shaft. The shear stress in the shaft due to torque is :

$$S_s = \frac{T'D_o}{2\,J} = \frac{16\,T'D_o}{\pi(D_o^4 - D_i^4)} = \frac{16\,T'}{\pi D_o^4(1 - \xi^4)} \tag{7.19}$$

where S_s = shear stress, lb/in^2 (assume the shear stress acceptable for high strength steel shafts is 10,000 lb/in^2)

J = polar moment of inertia, in^4

D_o = outer diameter, in

Solving equation (7.19) for the outer diameter (D_o) :

$$D_o = \left(\frac{16\,T'}{\pi\,S_s(1 - \xi^4)} \right)^{1/3} \tag{7.20}$$

The next step is to determine the critical shaft length. The critical shaft length for a simply supported tube is a function of the first critical speed (f_{ss}) :

$$f_{ss} = \frac{\pi}{2} \left(\frac{g\,E\,I}{L_{CS}^3\,W_S} \right)^{\frac{1}{2}} = \frac{\pi\,D_o}{8\,L_{CS}^2} \left(\frac{g\,E\,(1 - \xi^2)}{\rho_s} \right)^{\frac{1}{2}} \tag{7.21}$$

where f_{ss} = first critical speed, Hertz

g = acceleration due to gravity, 32.2 ft/sec^2

I = moment of inertia, in^4

L_{CS} = critical shaft length

E = modulus of elasticity, lb/in^2

W_S = shaft weight, lbs.

ρ_s = density of the material, ρ_s = 0.28 lb/in^3 for steel

The maximum speed of the shaft (n) is assumed to be $\leq 80\%$ of the first critical speed. Converting f_{ss} in equation (7.21) into RPM and solving for the critical shaft length :

$$L_{CS} = \frac{6\,\pi\,D_o}{n}\left(\frac{g\,E(1+\xi^2)}{\rho_s}\right)^{\frac{1}{2}} \qquad (7.22)$$

Once the critical length has been determined, it is easy to determine how many elements of length L_{CS} are needed in the particular length of shafting.

The weight of the shaft (W_S) can be calculated from :

$$W_S = \frac{\pi}{4}\rho_s D_o^2(1-\xi^2)L = 2.32\rho_s\left(\frac{T'}{S_s}\right)^{2/3}\frac{(1-\xi^2)L_{CS}}{(1-\xi^4)^{2/3}} \qquad (7.23)$$

where W_S = the weight of the shaft for critical length L_{CS}, lbs.

Multiplying W_S by the number of elements in the length of shafting gives the total weight of the shafting. Each shaft element consists of the shaft, a coupling (half on each end), and at least one bearing. The weight of the couplings is a function of torque :

$$W_C = 6.6 \times 10^{-4}\,T' \qquad (7.24)$$

where W_C = coupling weight, lbs.

T' = torque, in-lb.

In addition to the bearing, there will be lubrication and bearing supports. The weight of these components is classified as miscellaneous weight :

$$W_M = \frac{5\,D_o\,L_{CS}}{12} \qquad (7.25)$$

where W_M = miscellaneous weight, lbs.

D_o = outer diameter, inches

L_{CS} = critical length, inches

The transmission system also requires clutches in order to disconnect the engines and provide the necessary cross-connecting flexibility. Clutch weight is a function of torque and can be estimated from :

$$\text{Clutch Weight (lbs)} = 0.0045\, T' \text{ (Reference 15)} \qquad (7.26)$$

where T' = torque, in-lbs

These size and weight estimating equations will now be applied to the transmission system of the 5000 ton SES. The combined plant transmission will be analyzed first. Figures 20 and 21 provide a schematic and the transmission component nomenclature. The power from each FT9 engine will be split, with 31,000 H.P. going to the propeller and 7000 H.P. being transmitted forward to the four fans. The four fans on each side of the SES were arbitrarily spaced 20 feet apart. The after-most fan was spaced approximately 200 feet forward of the after perpendicular. The cross-connecting shaft was located approximately 10 feet forward of the gas turbines. The fan location, fan spacing, and propulsion engine location were held constant for both the combined and split palnts. The largest bevel gears in the transmission are the two double mesh gearboxes (BG1) between the engine and the propulsion planetary reduction gear. These bevel gears must transmit 38,000 H.P.. The 24-inch pitch diameter of these gears is at the upper limit of the current state-of-the-art gear manufacturing capability. The power splitting gearbox (BG3 and BG2) is also a double mesh bevel gear. The twin shafts which connect these two gearboxes (SP1) are five feet long and have an outside diameter of eight inches. Figure 22 shows that the bevel gear (BG2) which transmits power forward to the fans is vertical and forms a 20° angle with the inclined shaft of the propeller. This bevel gear (BG2) must be designed for 14,000 H.P. since, in the event

of an engine failure the gear must transmit the full fan horsepower. The bevel gear pair (BG4) on the engine side of the cross-connecting shaft (SI2) must also be designed for 14,000 H.P.. However, the single bevel gears (BG5) on the fan side of the cross- connecting shaft need only be sized for 7000 H.P..

The planetary reduction gears for the propellers and fans have reduction ratios of 9.6 : 1 and 2.93 : 1, respectively. The six clutches in the transmission system are located in such a way as to isolate the gas turbines in the event of a casualty, as-well-as to allow for cross-connecting the fan system. A summary of the transmission system is provided in Table 11. The total weight of the system is 140,620 pounds, representing a specific weight of 1.76 lb/H.P.. The shafting weight accounts for 31% of the total weight.

The split plant transmission system will be analyzed both for the four turbine and the five turbine cases. Looking first at the four turbine case, this plant uses two LM3500's for propulsive power and two GPTF990's for the lift power source. A schematic and the transmission component nomenclature is provided in Figures 23 and 24. This transmission system has far fewer components than the combined plant and is less complex. There are no bevel gearboxes and no long interconnecting shafts required in this arrangement. The planetary reduction gears for the propulsive system and the lift system have the same power and torque requirements as in the combined system. The reduction ratios are 9.6 : 1 for the propulsive reduction gear and 2.93 : 1 for the fan reduction gear. It was assumed that the GPTF990 and LM3500 gas turbines are available with only one direction of

rotation (driving the output shaft in a clockwise direction). In order to run the propellers in opposite directions an idler gear is needed to reverse the direction of rotation of one of the propeller shafts and one of the sets of fans. The formulas for sizing these idler gears are summarized in Reference 8. The nomenclature and terminology for these gears is provided in Figure 25. The method used to estimate the reduction gear weights is well known and is called the Dudley Q-factor method (Reference 15). This method is based on the assumption that gear weight is a function of center distance (C), and face width (F). The reference volume for a single reduction gear set is then equal to the center distance squared times the face width of the gear. The Q-factor is a result of including the surface durability factor (K-factor) in the reference volume. The reference volume is :

$$C^2F = \left(\frac{d_p + d_g}{2}\right)^2 F = \frac{d_p^{\,2}}{4} F\,(M_g + 1)^2 \qquad (7.27)$$

where C = centerline distance between the gear and the pinion, in.

F = face width of gears, in.

d_p = pinion pitch diameter, in.

d_g = gear pitch diameter, in.

Substituting the K-factor to eliminate the face width (F) from the right side of equation (7.27) :

$$C^2F = \frac{\overline{W}\,d_p(M_g + 1)^3}{4\,K\,M_g} \qquad (7.28)$$

where $\overline{W} = \dfrac{2\,T'}{d_p}$ = tangential force between meshing gears, lb.

$T' = \dfrac{(12)(33{,}000)P}{2\,\pi\,n_p}$ = input torque, in-lb.

P = power being transmitted, H.P.

n_p = speed of rotation of pinion, RPM

K = surface durability factor, lb/in^2

Substituting the equation for torque and tangential force into equation (7.28) gives :

$$C^2F = \frac{3.15 \times 10^4}{K} \left(\frac{P(M_g + 1)^3}{n_p M_g} \right) \text{ in}^3 \qquad (7.29)$$

The Q-factor has been defined by Dudley as the group of terms in the brackets :

$$Q_D = \frac{P(M_g + 1)^3}{n_p M_g} \text{ H.P./RPM}$$

The weight of single reduction gears including gear casing, oil reservoir, and pump is given as :

$$w_s = 2.9 \times 10^4 (Q_D/K)^{0.8} \text{ (Standard gears)} \qquad (7.30a)$$

$$w_s = 1.6 \times 10^4 (Q_D/K)^{0.9} \text{ (Lightweight gears)} \qquad (7.30b)$$

where w_s = weight of gear, lbs.

Since the reversing idler gears are not being used to change the RPM of the output, the reduction ratio (M_g) is 1 : 1. It is assumed that $d_p = d_g$, the K-factor is 500, and $F = 0.70\ d_p$ (Reference 16). Combining these assumptions and the previous equations, the reversing idler gear sizes and weights were calculated for both the propulsive and lift system. Table 12 summarizes the size and weight of the four turbine split plant. The total weight of the transmission is 64,688 pounds.

The five turbine split plant has the same propulsive gas turbines, propulsive planetary reduction gear, and propulsive reversing idler gear as the four turbine plant. However, the three lift fan gas turbines re-

quire a system of bevel gears and clutches to provide power to the lift fans. A schematic showing the layout and nomenclature of the five turbine plant transmission system is provided in Figures 26 and 27. The LM500 gas turbines are operated at 4667 H.P. and 6500 RPM. The bevel gears, shafting, clutches, and fan reduction gears were sized using the equations developed in the combined plant analysis. The bevel gearboxes were all designed to transmit 9333 H.P. to account for an unbalanced fan load. A summary of the size and weight of the transmission components in the five turbine split plant is provided in Table 13. The weight of the entire transmission system is 78,038 lbs.

8. WEIGHT ESTIMATES OF MISCELLANEOUS PLANT COMPONENTS

In previous sections the engines, reduction gears, fans, transmission shafts, and clutches have been analyzed and the weight and size of these equipments have been analyzed. The complete power plant system also includes several other components whose weight must be estimated in order to arrive at the total plant weight. These components are :

1. Propellers
2. Propeller Shafts
3. Fan Ducting
4. Gas Turbine Inlet and Exhaust Ducting
5. Seal Ducting

The first components whose weights will be estimated are the two partially submerged supercavitating controllable reversible pitch propellers. The weight estimating correlation is taken from Reference 17 :

$$W_P = 6.5\,D_P^{\,3} \quad \text{(Fixed pitch propeller)} \tag{8.1a}$$

$$W_P = 8.0\,D_P^{\,3} \quad \text{(CRP propeller)} \tag{8.1b}$$

where W_P = weight of titanium propeller, lbs.

D_P = propeller diameter, ft.

Using a propeller diameter of 13.5 feet, the weight of each propeller is 19,689 lbs.

The propeller shaft for the 5000 ton SES will be designed using the U.S. Navy criteria described in Reference 18. The shafting will be designed to withstand a 20% overload of full power torque. The horsepower requirement is for 31,000 H.P. per shaft. The material selected for the shafting

is titanium (6Al-4V) with the following properties :

Yield Point (Y.P.)	=	120,000 lb/in^2
Failure Level (F.L.)	=	130,000 lb/in^2
Density	=	277 lb/ft^3

For this analysis it was assumed that the ratio of inner diameter (d_i) to outer diameter (d_o) is $d_i/d_o = 2/3$. A summary of the equations used in this analysis is provided in Table 14. The methodology used to determine the shafting size was to calculate the resultant steady stress $(S_R)_{ss}$ and the resultant alternating stress $(S_R)_{alt}$ in terms of the inner and outer diameter where $d_i = 2/3\ d_o$. These terms are then substituted into equation (8.11). Values of d_o are then selected and equations (8.6), (8.8), and (8.11) are solved until the shaft diameter satisfies the design criteria with a factor of safety (F.S.) equal to 2.00. Applying the equations of Table 14, the shaft dimensions are :

$$d_o = 20 \text{ inches}$$

$$d_i = 13.3 \text{ inches}$$

The shaft length is shown in Figure 28 to be 40.97 feet. The volume of the shaft material is :

$$\text{Vol.} = \frac{\pi}{4}\left(d_o^{\,2} - d_i^{\,2}\right) \text{ X Length} \tag{8.15}$$

$$\text{Vol.} = 49.66 \text{ ft}^3$$

The shaft weight is :

$$\text{Shaft Weight} = \text{Vol. X Density} \tag{8.16}$$

$$\text{Shaft Weight} = 13{,}756 \text{ lbs.}$$

In addition to the shaft itself, the weight of the bearings and couplings must be considered. It has been assumed that only one coupling is required.

The coupling weight is calculated using equation (7.24) :

$$W_C = 6.6 \times 10^{-4} \, T'$$

where W_C = coupling and bearing weight, lbs.

T' = torque, in-lb

Using the values of propeller torque from Table 3 :

$$W_C = 1026 \text{ lbs/shaft}$$

The weight of the lubrication and bearing supports is calculated using equation (7.25) :

$$W_M = 4100 \text{ lbs.}$$

The weight estimates of the gas turbine inlet and exhaust ducting is based on the following weight assumptions :

	Large Turbines 5000 H.P. (lb/ft^2 of duct surface area)	Small Turbines 5000 H.P. (lb/ft^2)
Inlet (Square)	6	4
Exhaust (Round)	7	5

The average inlet duct gas velocities are assumed to be 70 ft/sec, the average exhaust velocity is assumed to be 150 ft/sec, and engine cooling air is about 10% of the combustion air requirements. The mass flow rate of the combustion air is determined from engine data. The surface area for the ducting is calculated from :

$$S.A. = 4 \, A_D^{\frac{1}{2}} \, L \qquad \text{(Inlets)} \qquad (8.17a)$$

$$S.A. = 2 \left(\frac{A_D}{\pi}\right)^{\frac{1}{2}} \pi \, L_D \qquad \text{(Exhausts)} \qquad (8.17b)$$

where S.A. = surface area, ft^2

A_D = cross sectional area, ft^2

L_D = length, ft.

The actual location of the gas turbines and fans is shown in Figure 29. Inlet and exhaust ducting weight estimates are summarized in Table 15.

Figure 29 also shows the location of the SES fan ducts. In this analysis it will be assumed that all fans are located at the wet deck level and exhaust directly into the cushion plenum. The fan inlet ducting is assumed to be three decks (27 feet) high. The air inlet velocity was assumed to be 6000 ft/min. with a mass flow rate of 4050 ft^3/sec. An inlet duct diameter of 6.25 feet was selected and the weight of the duct per unit of surface area of 4 lbs/ft^2 was selected. The surface area was calculated using equation (8.18b). The fan ducting weight was then estimated to be 3313 lbs for each duct. In addition to the fan inlet duct, there must be a provision for ducting to provide air pressure to the forward and after cushion seals. From Figure 29, it can be seen that a duct is provided from the forward and after fans on each side of the SES to the forward/after cushion seal. The total length of the duct is 390 feet. A duct diameter of 6.25 feet was selected. The total ducting weight of 15,315 lbs was calculated for these ducts. A summary of these weights is provided in Table 15.

9. COMPARISON OF THE SPLIT AND COMBINED PLANTS

In previous sections, the total weight of the power plant and transmission system for three plant configurations was estimated. A summary of these weight estimates is provided in Table 19. The range of the three plant configurations was calculated using the methodology outlined in Chapter 2. The results for the three plants (combined, four turbine split, and five turbine split) for a 25% fuel fraction are shown in Tables 20 - 43. The results are also summarized graphically in Figure 30. Sample calculations are provided in Appendix A.

The results show that the combined plant yields a greater range than the split plant configurations. The combined plant with its heavier and more complex transmission carries less fuel than the split plant, but achieves a greater range due to the fuel economies gained from operating both the lift and propulsion systems with the same prime mover.

Figure 30 also demonstrates the difference between the static lift dominated lift system and the wave pumping action dominated lift system which takes effect at speeds greater than 46 knots. Below 46 knots it is possible to reduce the fan horsepower as the SES consumes fuel and thus becomes lighter. This results in increased range due to the lower fuel rate. Above 46 knots, the action of the waves sweeping out the air in the cushion dominates the static lift requirements. Cushion horsepower then varies as a function of speed. The range of all plant types decreases as the ship speed increases. Figure 30 shows that there is a discontinuity at 46 knots, beyond which the range of all plant types decreases as the SES speed increases. For purposes of comparison, the static lift trend line

was extrapolated to a speed of 50 knots to clearly show this effect.

10. RECOMMENDATIONS

While the results of this analysis favor the combined plant, several other design studies should be conducted in order to provide the operator with a summary of the advantages and disadvantages of the alternate plant configurations. First, a reliability analysis of both plant types should be performed. This is especially important for the combined plant since that arrangement has the most complicated transmission system of the three plants discussed. Due to the added complexity, the combined plant may require additional maintenance personnel and additional repair parts.

Another factor which should be analyzed is the effect of alternative plant arrangements on the weight of both the combined and split plants. For instance, if the fans in the combined plant were relocated closer to the FT9 gas turbines, a large section of the interconnecting transmission shaft could be eliminated. Since this section of shafting (SI3) makes up nearly 5% of the total plant weight, the weight savings could be significant. A reduction in the shafting weight would make the combined plant even more attractive since the fuel weight could be increased, thus increasing the range of the combined plant SES.

There is also a trade-off between the added sidehull drag and the reduced transmission system complexity which would result from widening the sidehulls. This would allow the reduction gear and possibly the engines to be located closer to the propulsors. If the engines were mounted in the sidehulls, the technical problem of operating the turbines at a 20° angle of inclination would be avoided. The 20° angle of inclination exceeds the operating parameters for the modern gas turbines used in this analysis.

However, the gas turbines, propulsion fans, and main gearbox in the VT.2, an amphibious hovercraft built by Vosper - Thornycroft, are mounted at an angle of 13.5° to the horizontal (Reference 19).

Any further comparisons of the three plant configurations should also investigate the amount of usable volume required by each plant. For instance, the long interconnecting and cross-connecting transmission shafts in the combined plant require some type of safety shrouding or a tunnel, and thus may have a severe impact on usable volume and access on the wet deck level.

One of the key assumptions used in this comparison was that the speed of the SES was kept constant. An important extension of this study would be to calculate the ship speed which results in the lowest fuel flow rate (lbs/hr) at specific gross weight conditions. These values could then be used to determine the maximum endurance of the SES by integrating between the maximum ship gross weight and the empty weight condition.

Another interesting study would be to apply the methodology used in this analysis to compare a combined and a split plant which use waterjet propulsion.

Since this analysis was primarily concerned with feasible rather than optimal configurations, it would be informative if an analysis could attempt to optimize a water screw propelled 5000 ton SES. Many of the parameters and relationships necessary to optimize this system have been discussed here.

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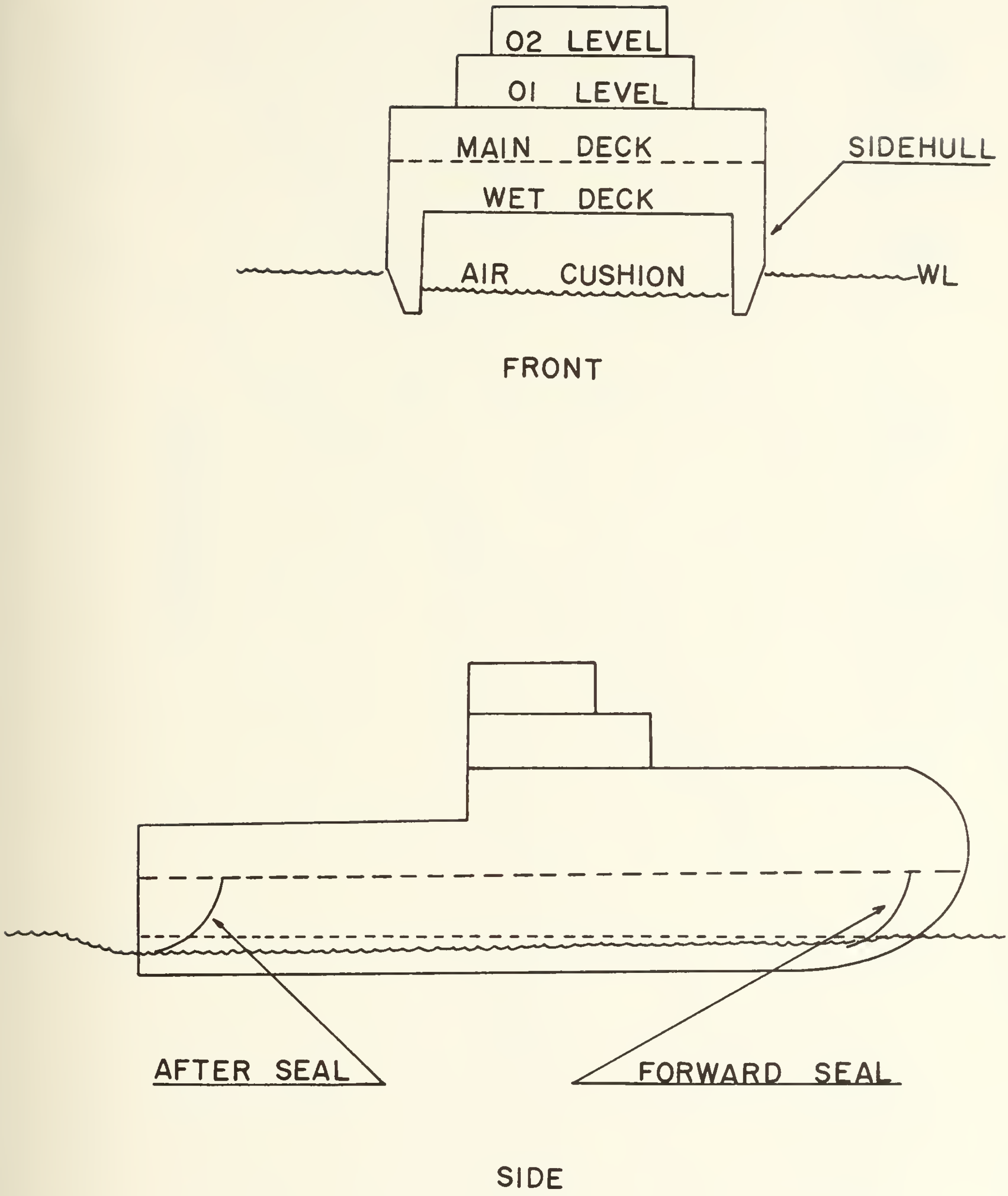


FIGURE 1 SES Terminology

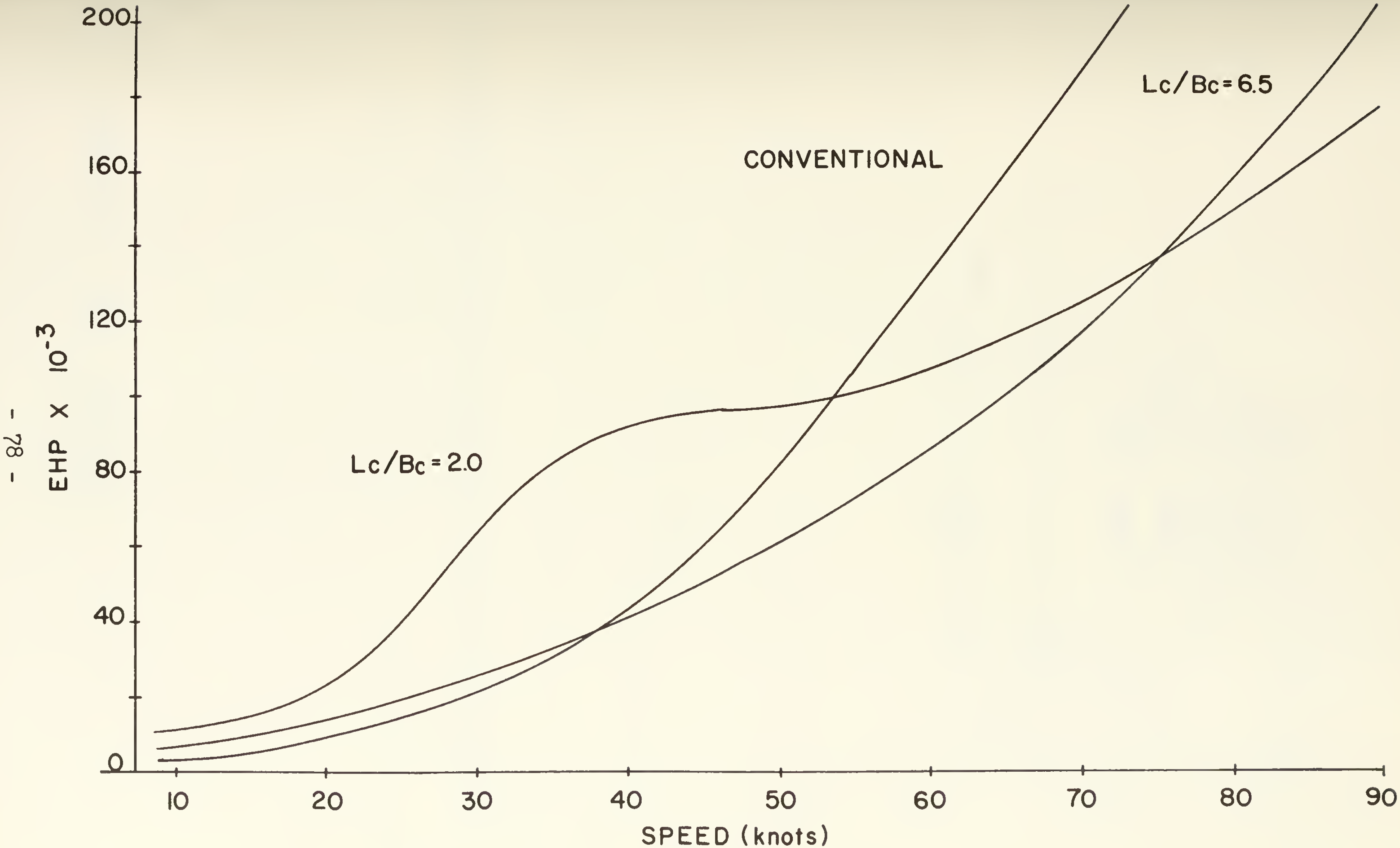


FIGURE 2 Characteristic Powering Relationships

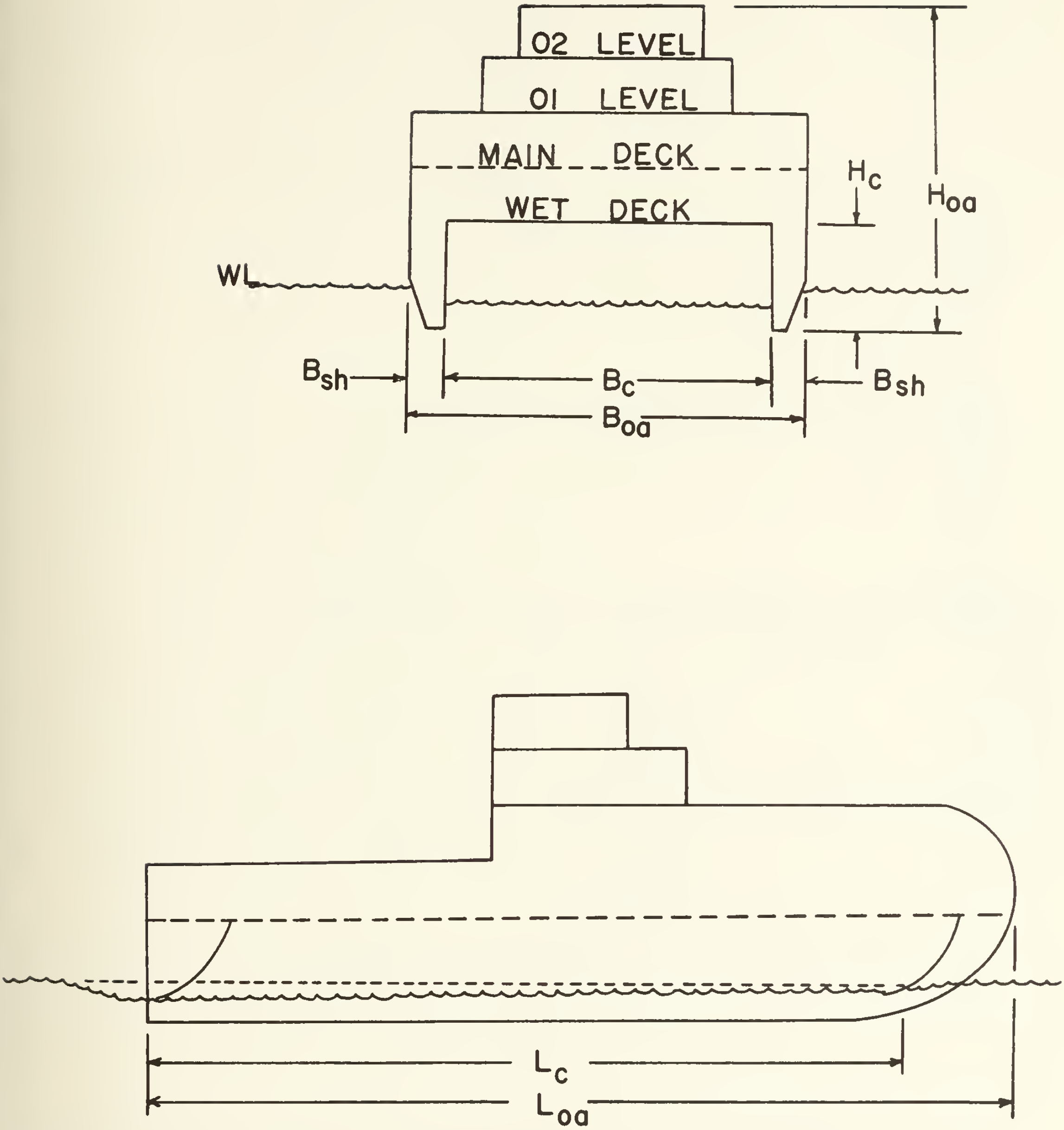


FIGURE 3 SES Geometry

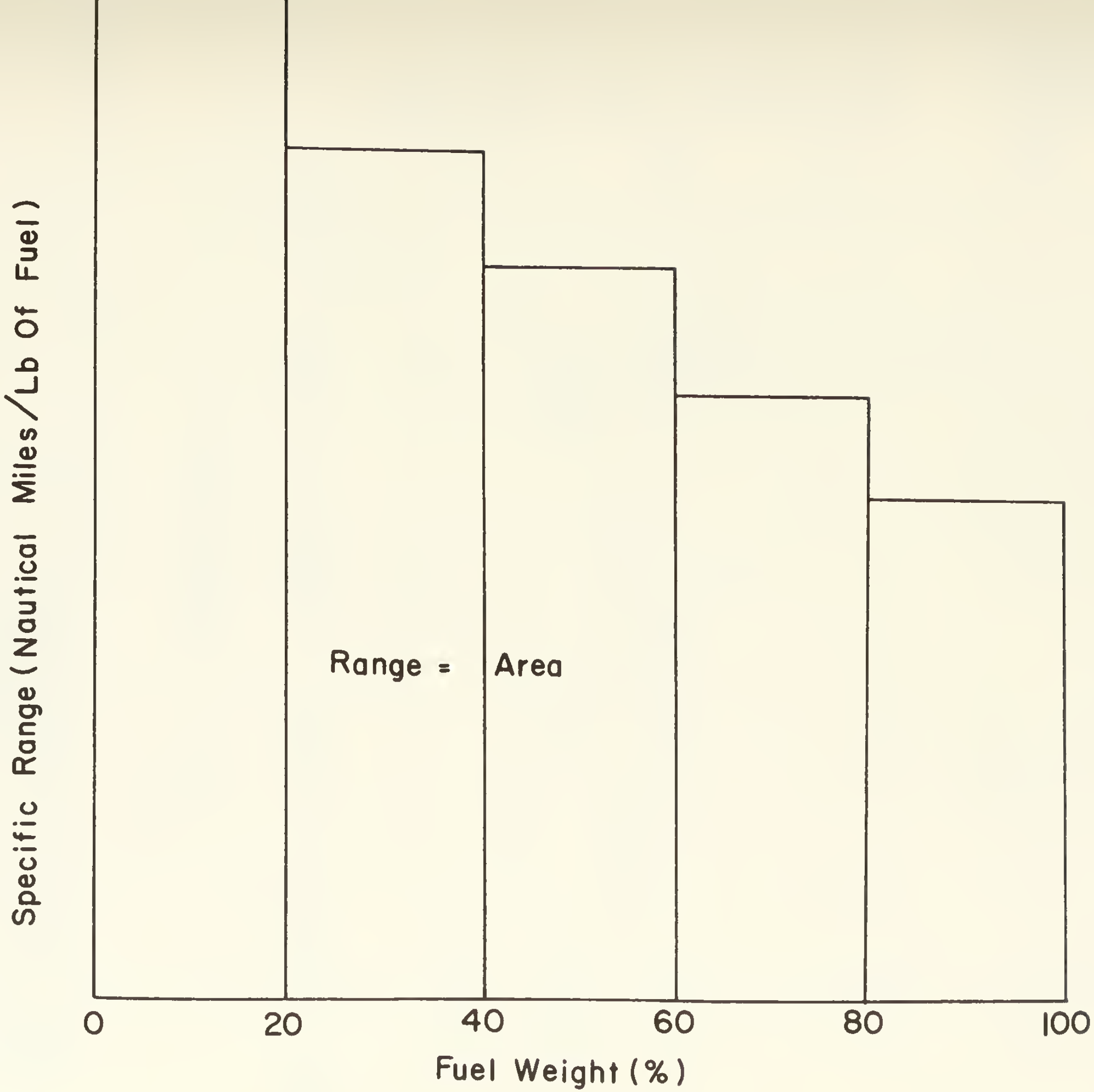


FIGURE 4 Range Integration Methodology

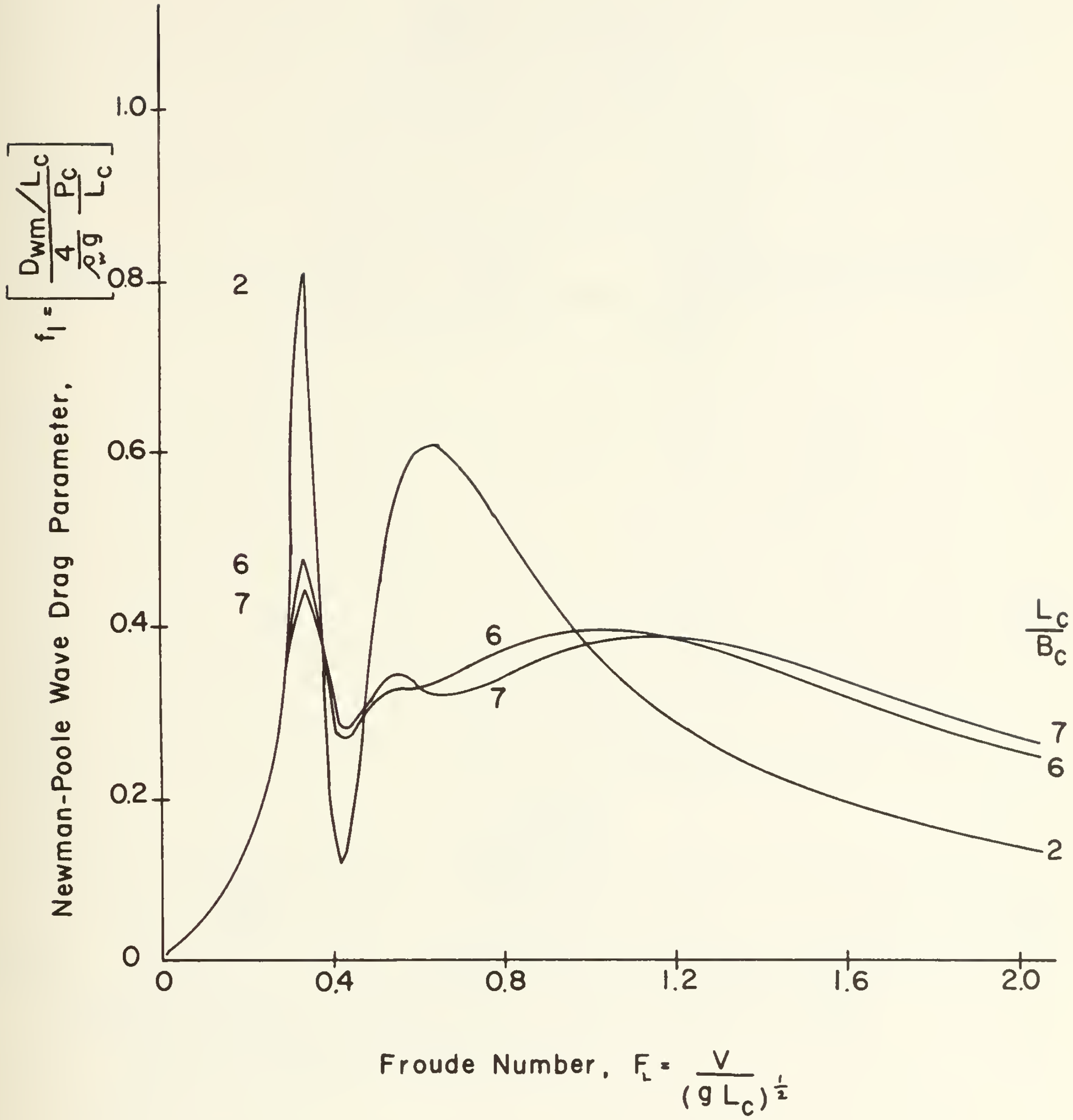
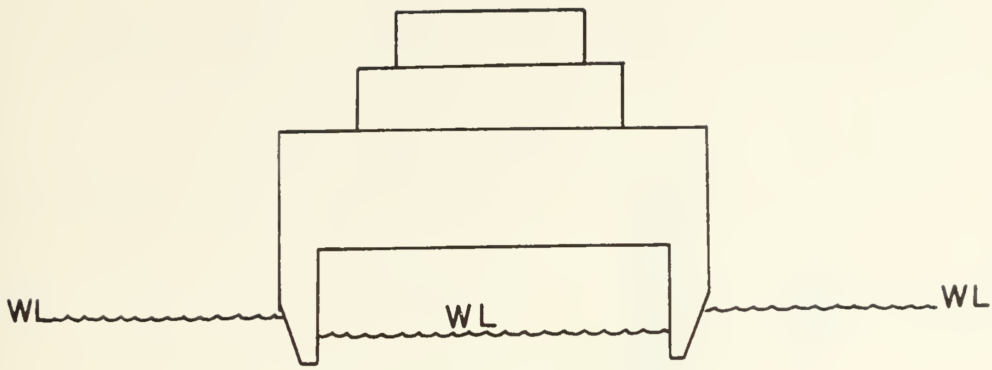


FIGURE 5 NEWMAN-POOLE WAVE DRAG PARAMETER VERSUS FROUDE NUMBER



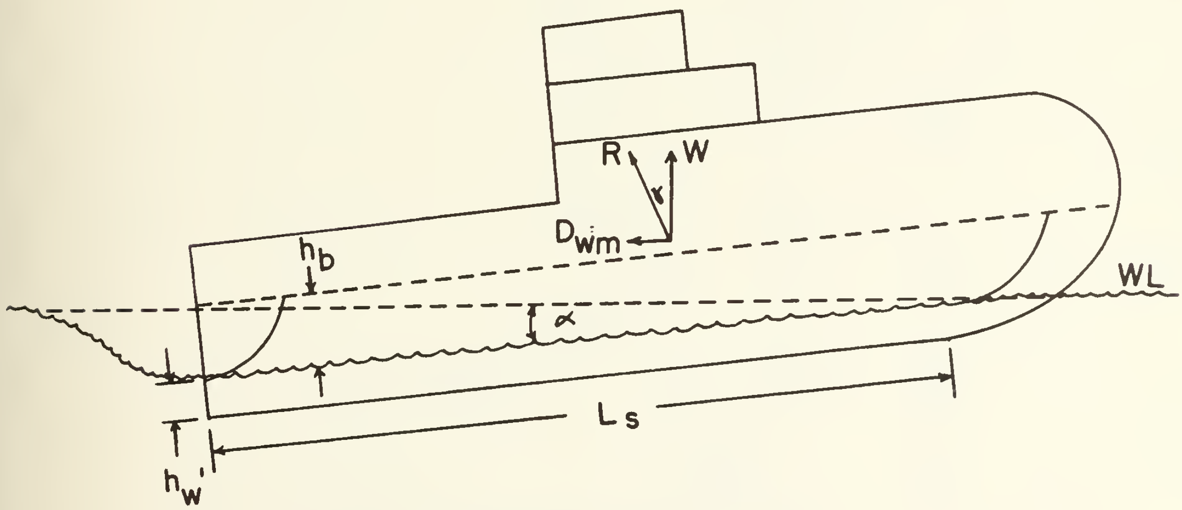


FIGURE 5a SES Sidehull Wetted Surface Geometry

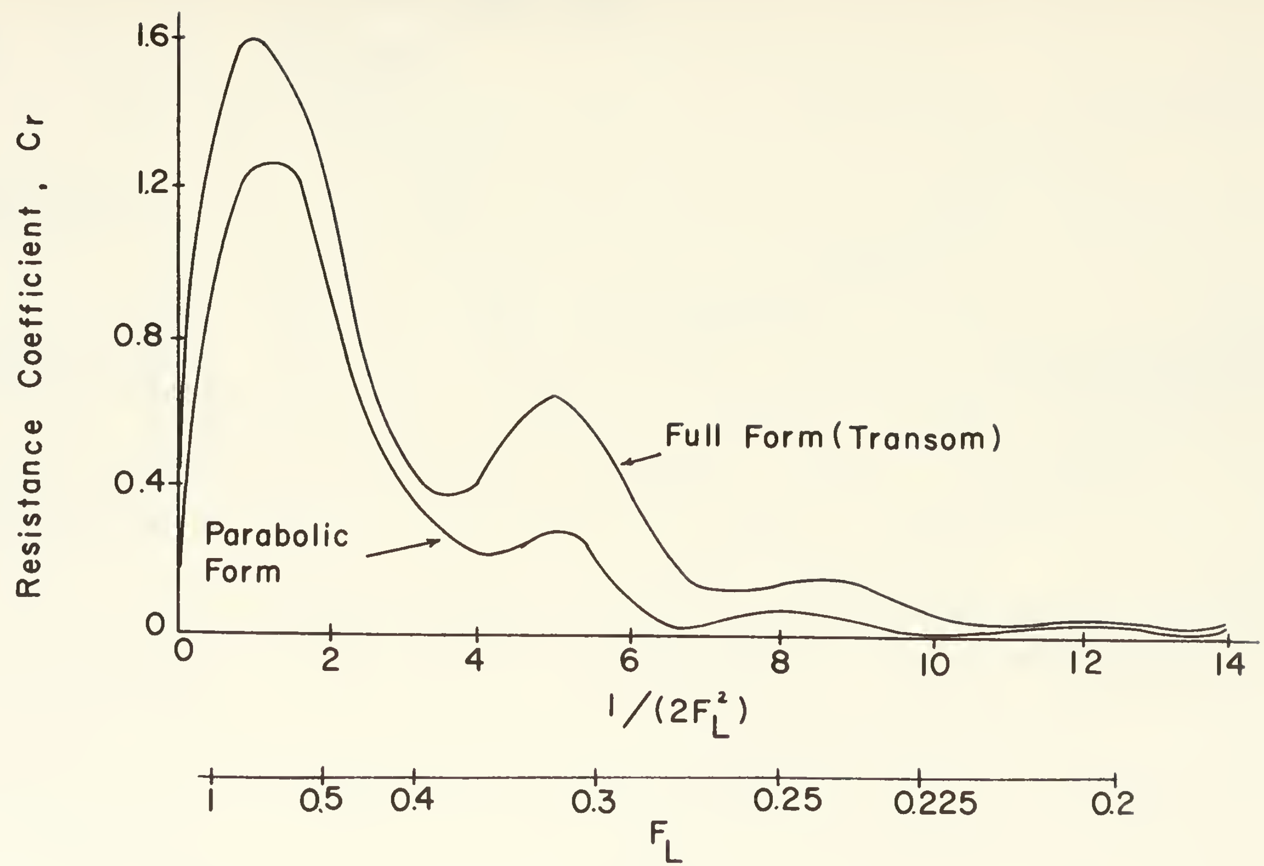


FIGURE 6 Wave Resistance Coefficient For Thin Bodies

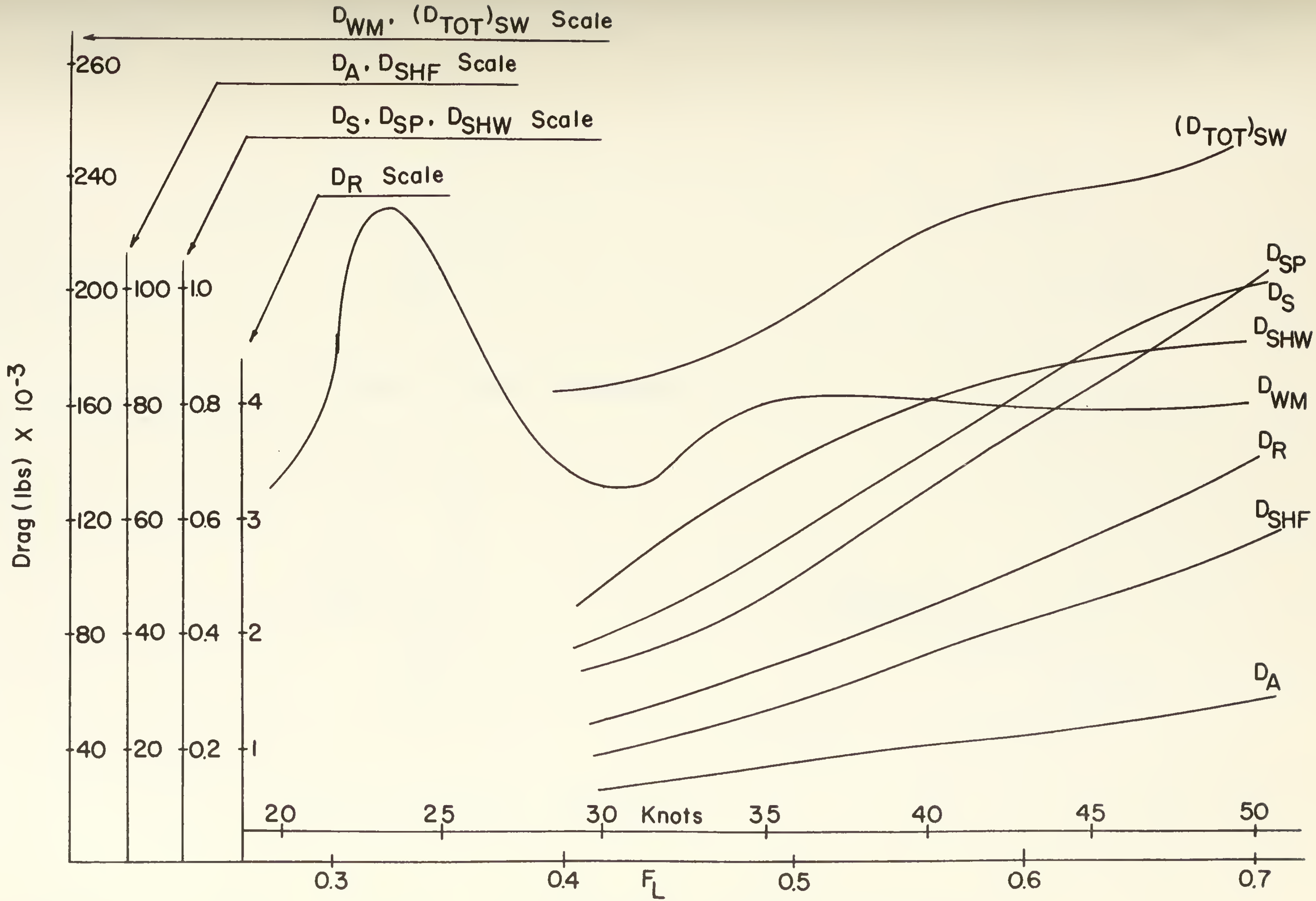


FIGURE 7 SES Drag Summary

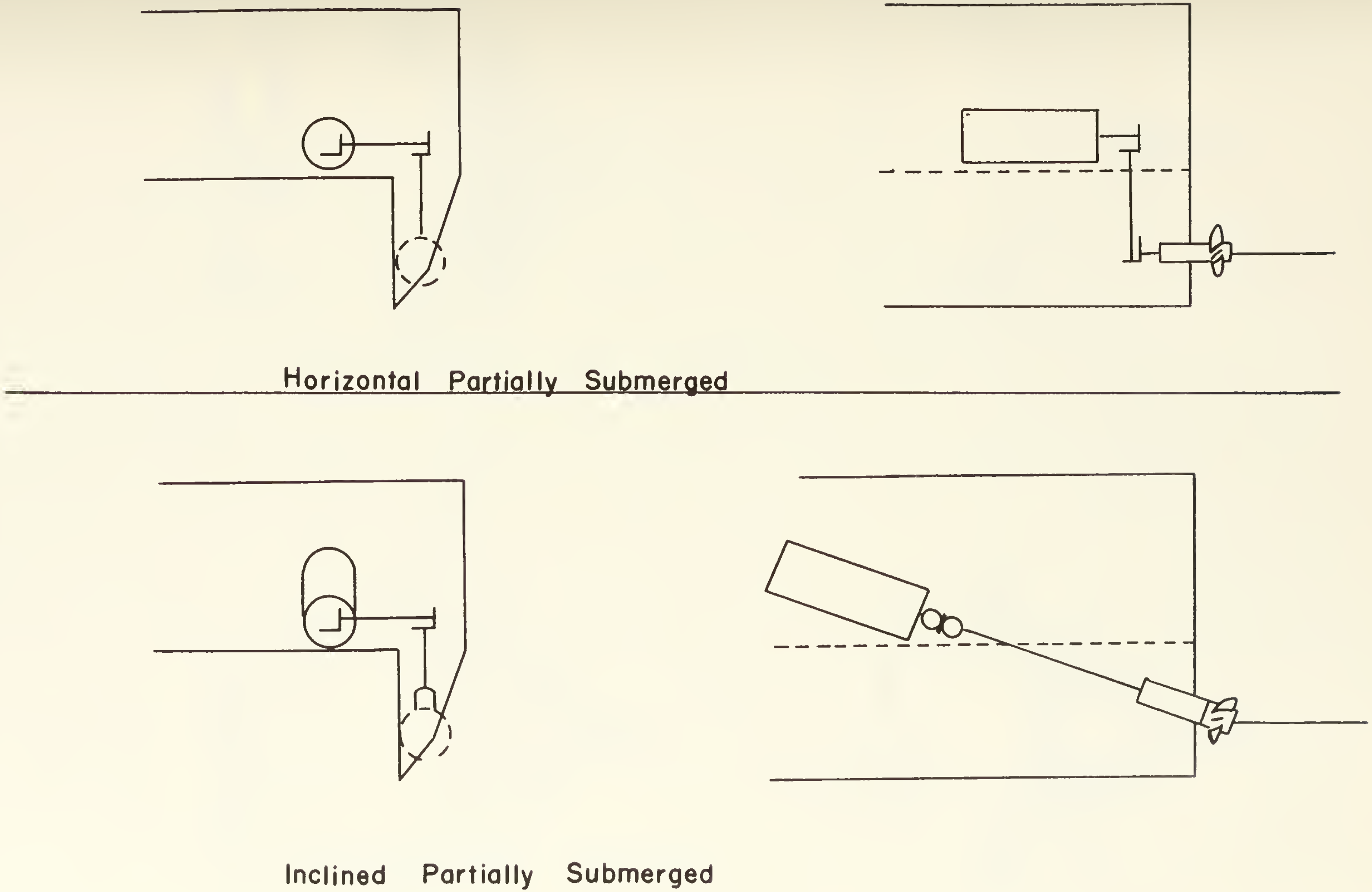
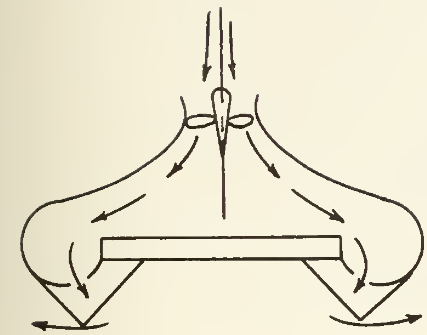
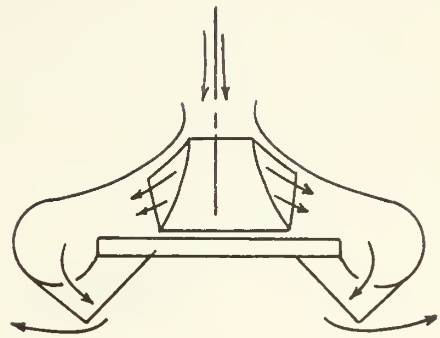


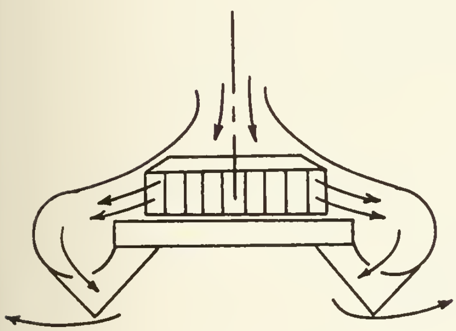
FIGURE 8 Propeller Mounting Concepts For A SES



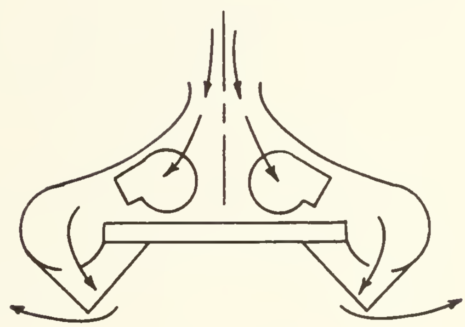
(A) Axial



(B) Mixed Flow

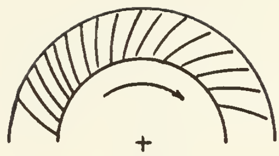


(C) Centrifugal, Axis
Parallel To Inlet

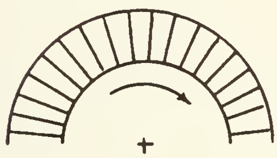


(D) Centrifugal, Axis
Perpendicular To Inlet

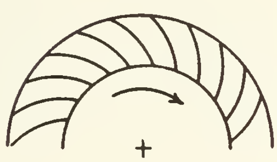
FIGURE 9 SES Fan Concepts



(A) Forwardly Curved Blades



(B) Radial Blades



(C) Backwardly Curved Blades

FIGURE 10 Blade Curvature Of Centrifugal Fans

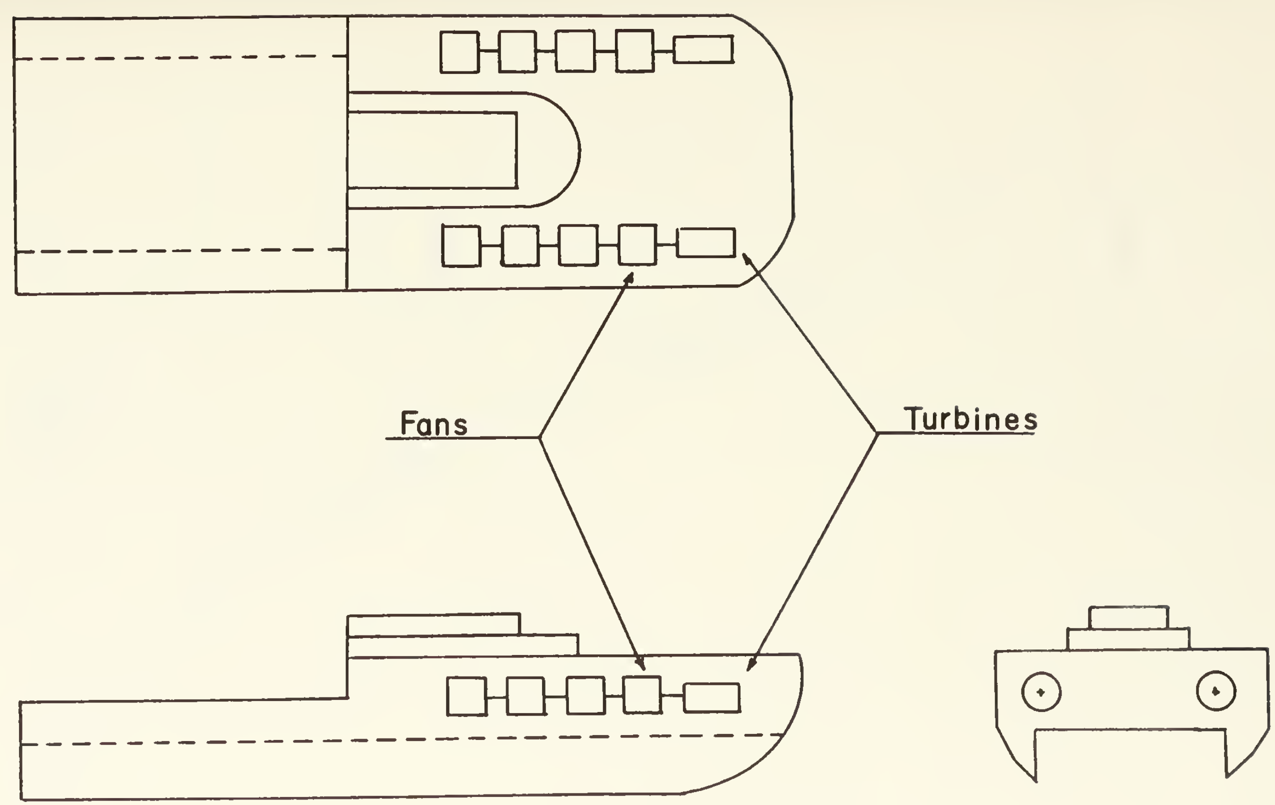


FIGURE II Centrifugal Fan Installation Concept

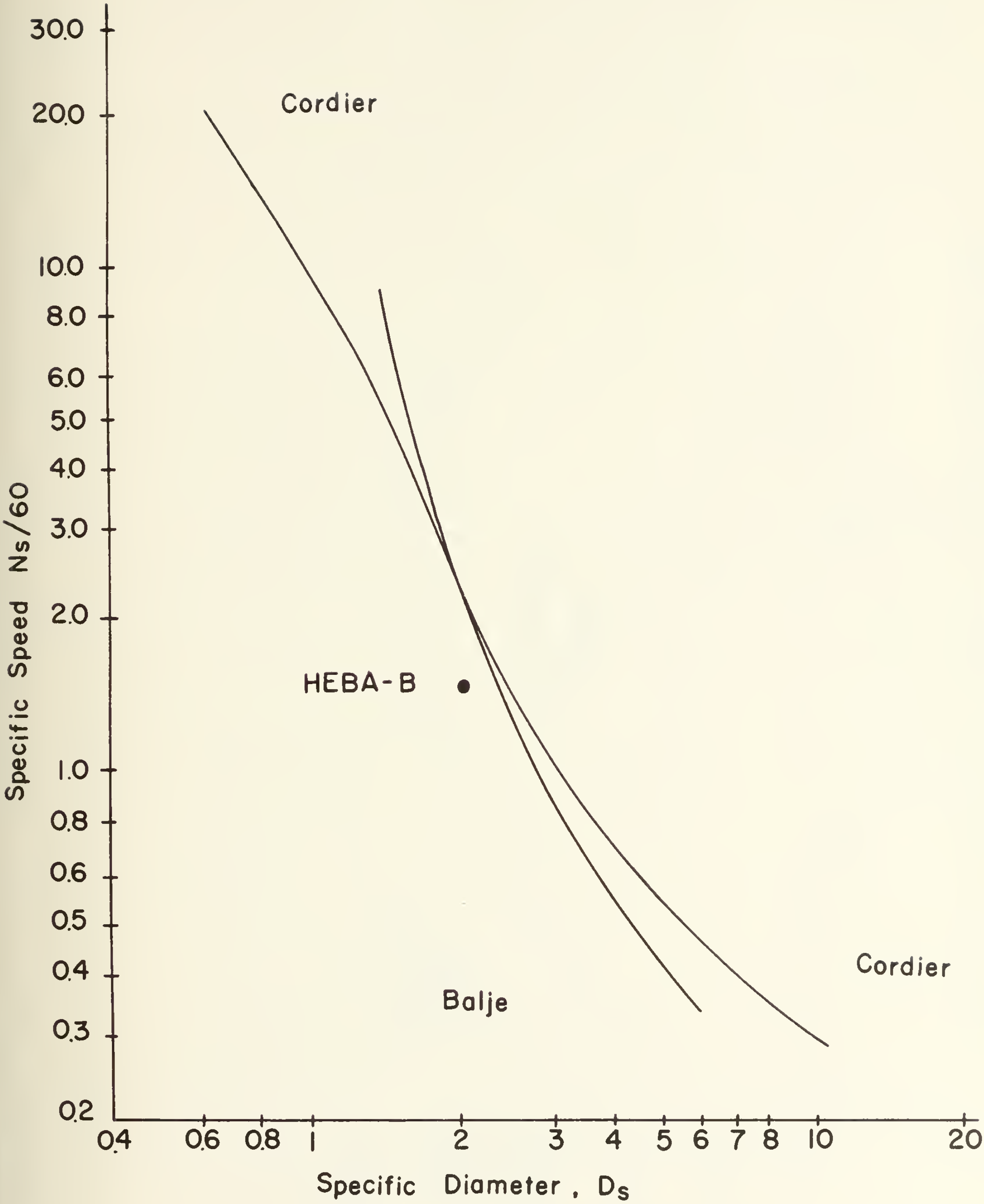


FIGURE 12 Comparison of Cordier and Balje Results

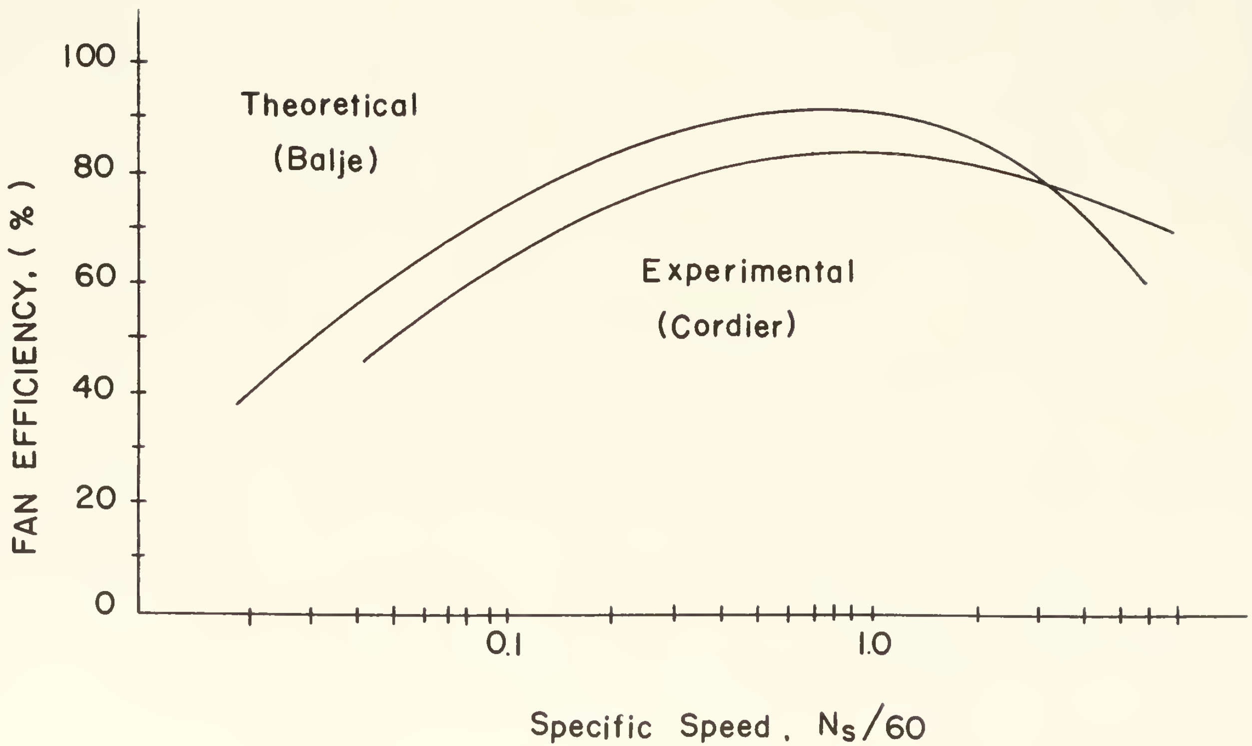
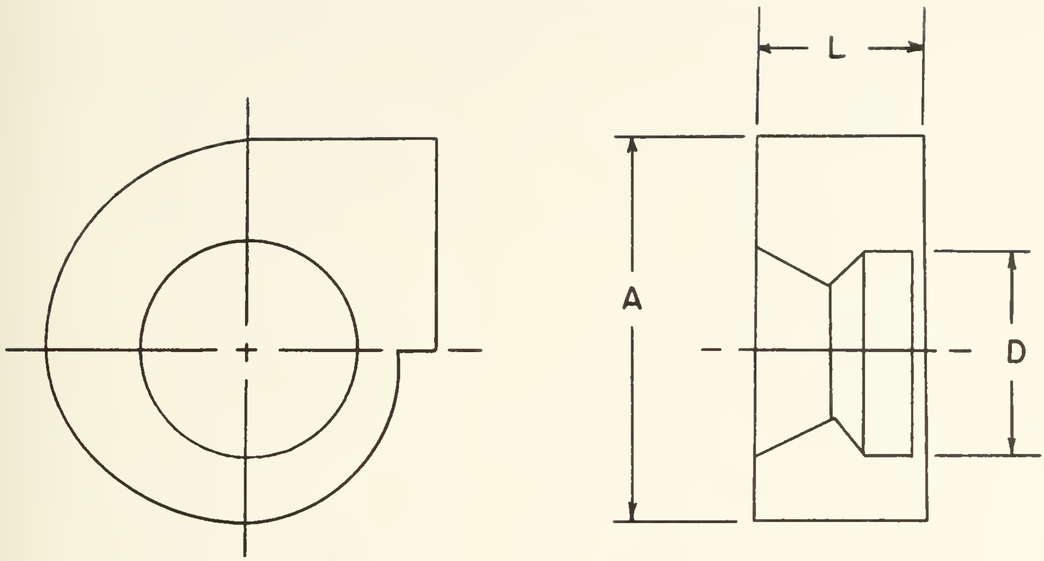


FIGURE 13 Maximum Efficiency Versus Specific Speed



L = 0.80D

A = 1.87D

FIGURE 14 HEBA-B Fan Dimensions

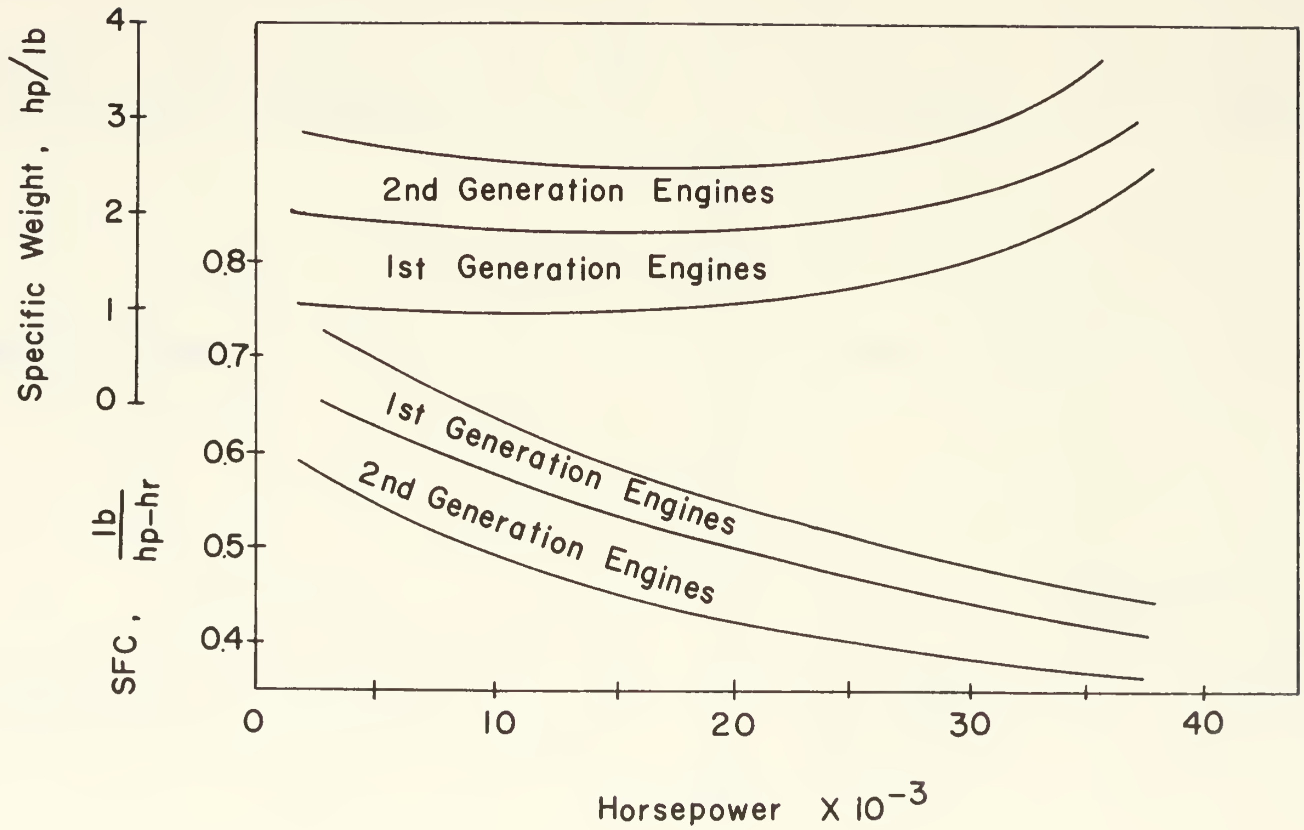


FIGURE 15 SFC and Specific Weight Versus Power

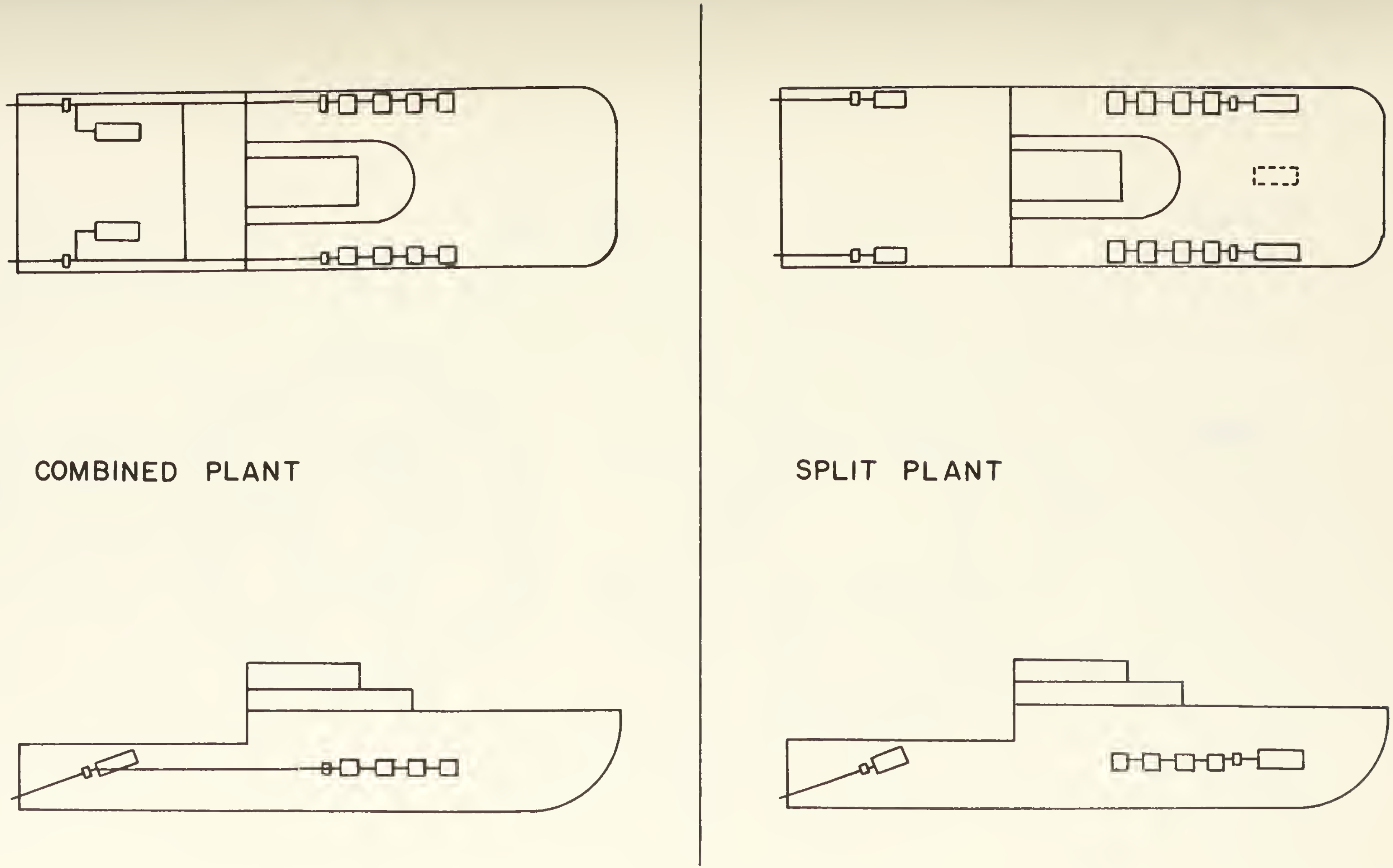


FIGURE 16 PROPOSED SES TRANSMISSION SYSTEMS

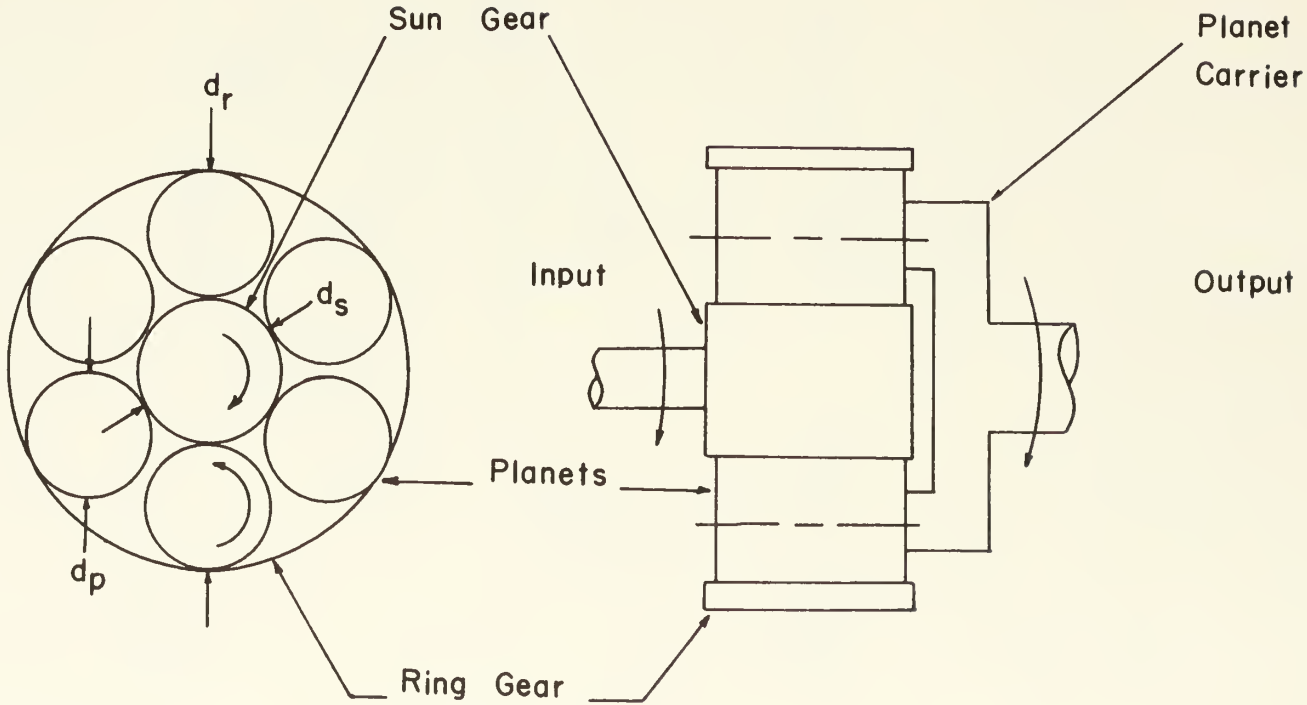


FIGURE 17 Planetary Reduction Gear Nomenclature

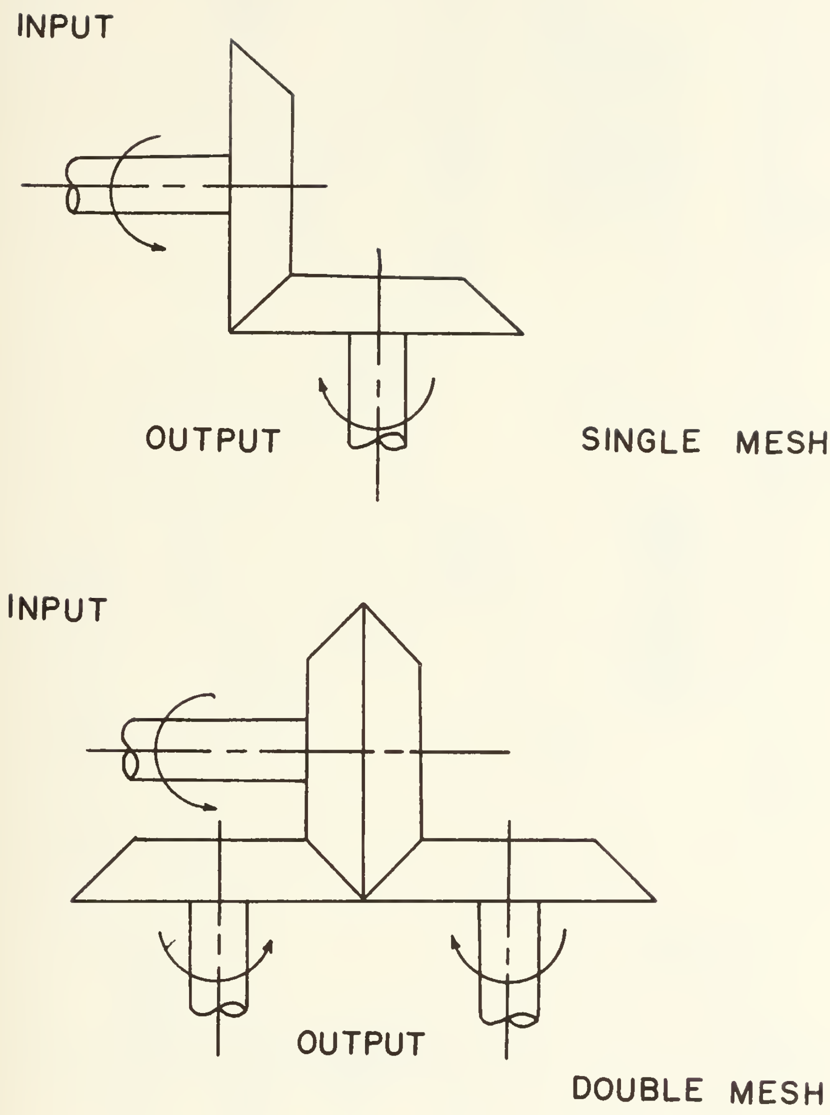


FIGURE 18 BEVEL REDUCTION GEAR ILLUSTRATION

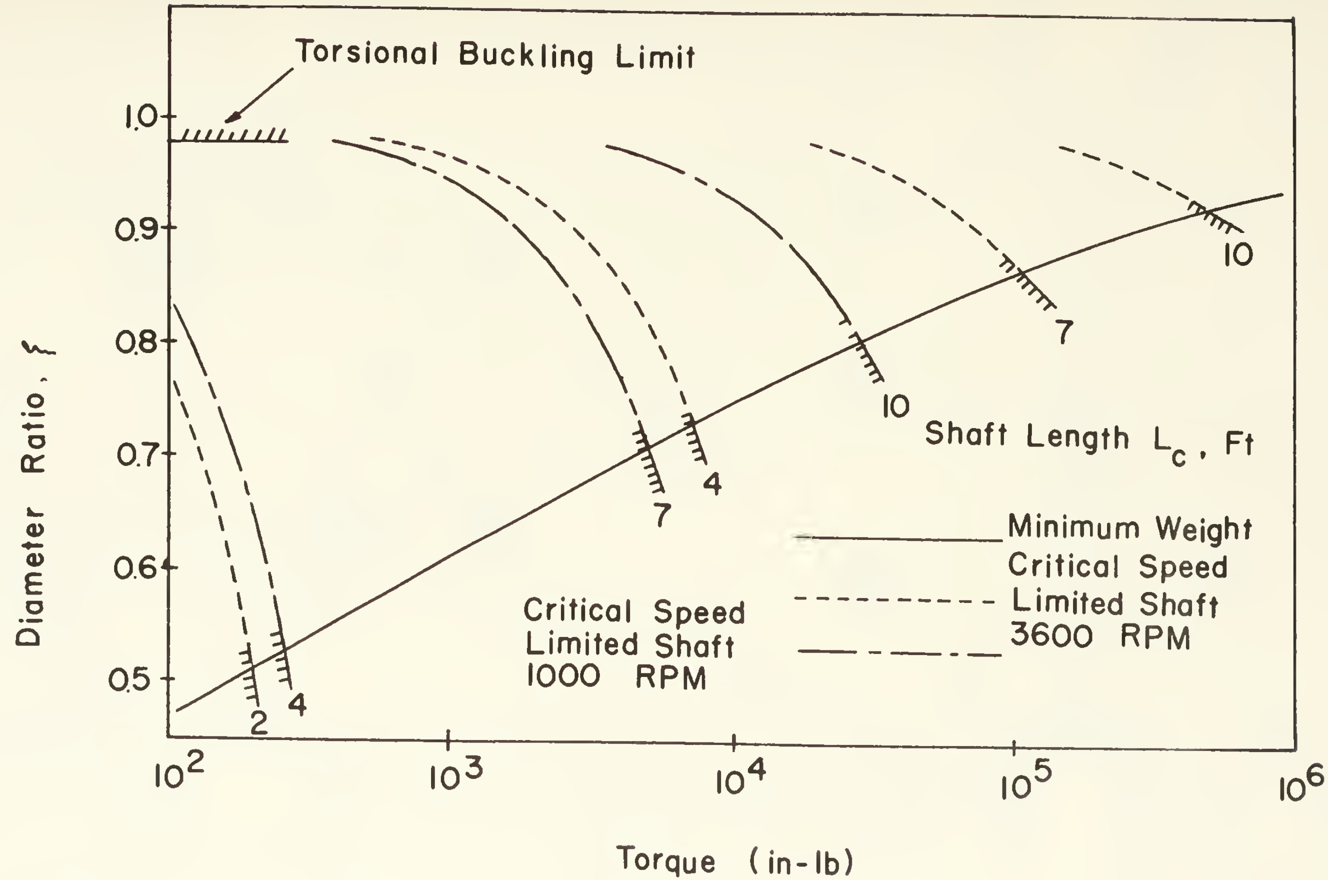


FIGURE 19 Diameter Ratio for Minimum Weight Shaft

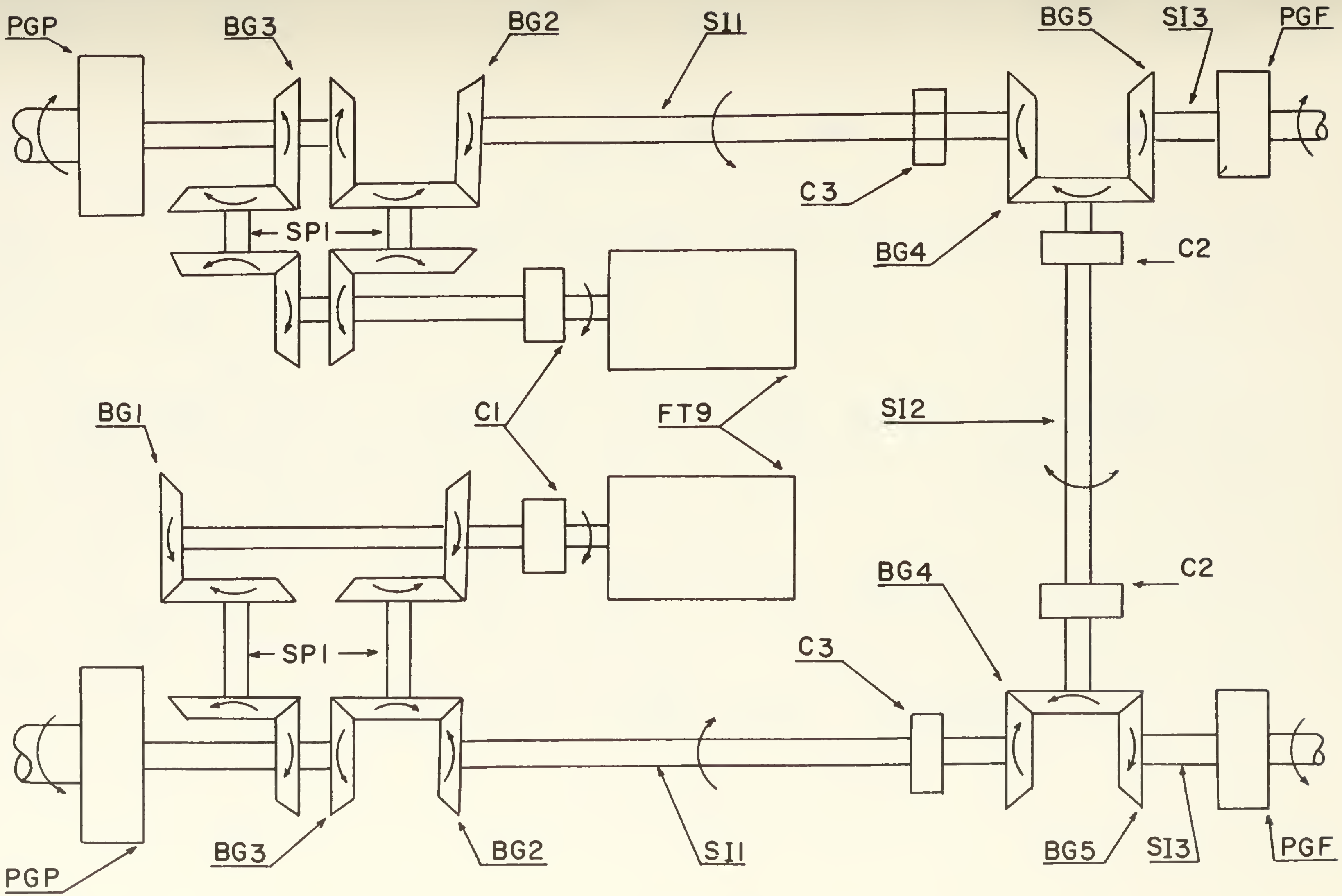


FIGURE 20 COMBINED PLANT TRANSMISSION SCHEMATIC

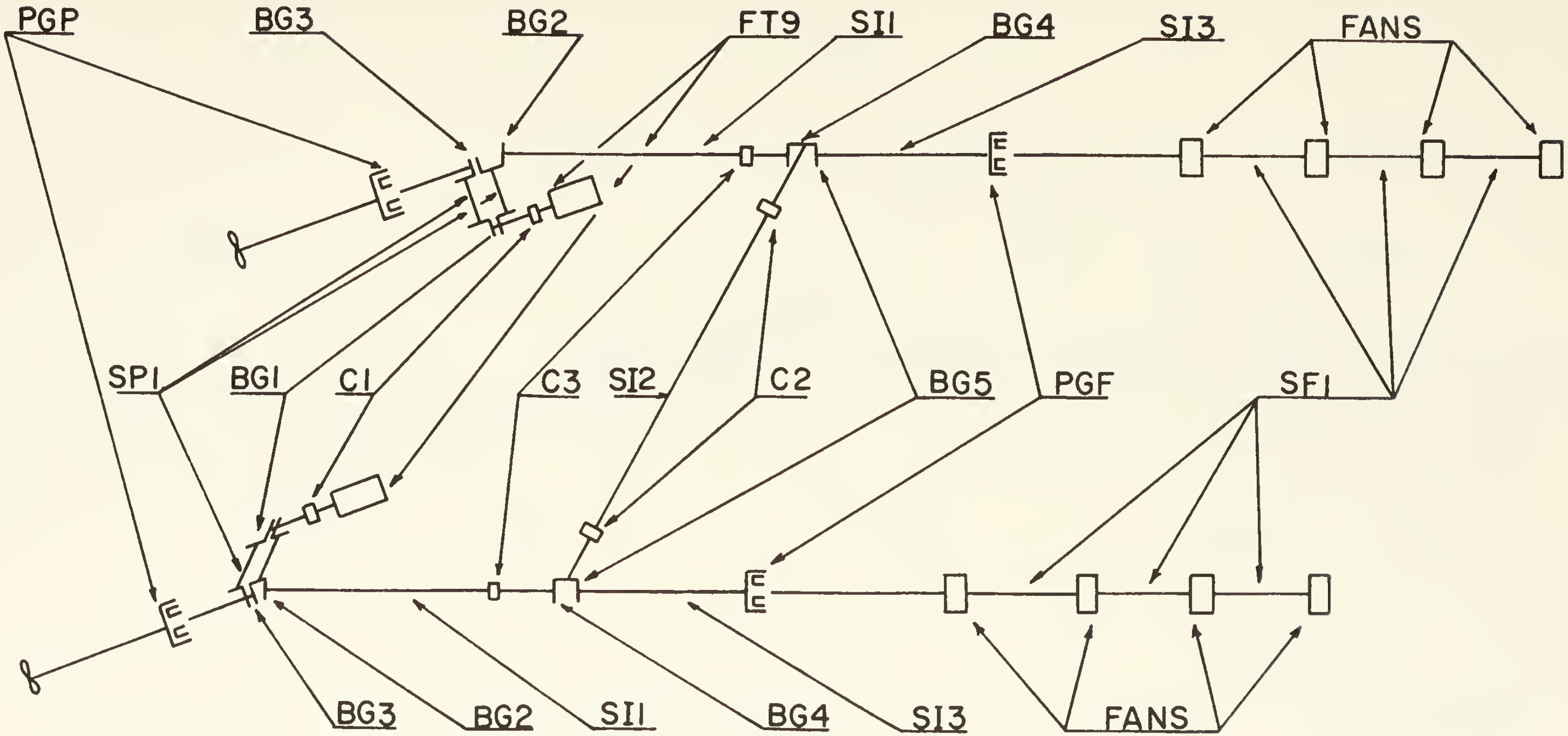
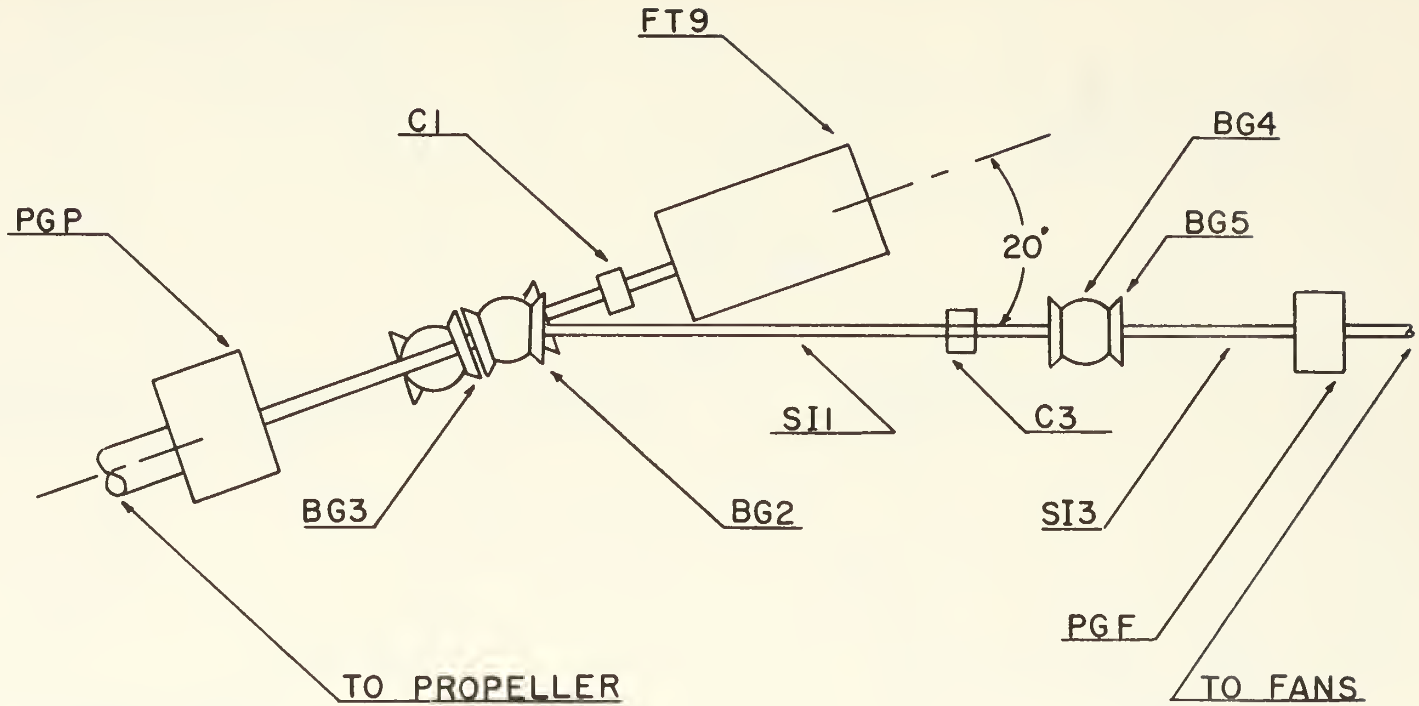


FIGURE 21 COMBINED PLANT TRANSMISSION DIAGRAM



SIDE VIEW

FIGURE 22 POWER SPLITTING GEARBOX SCHEMATIC

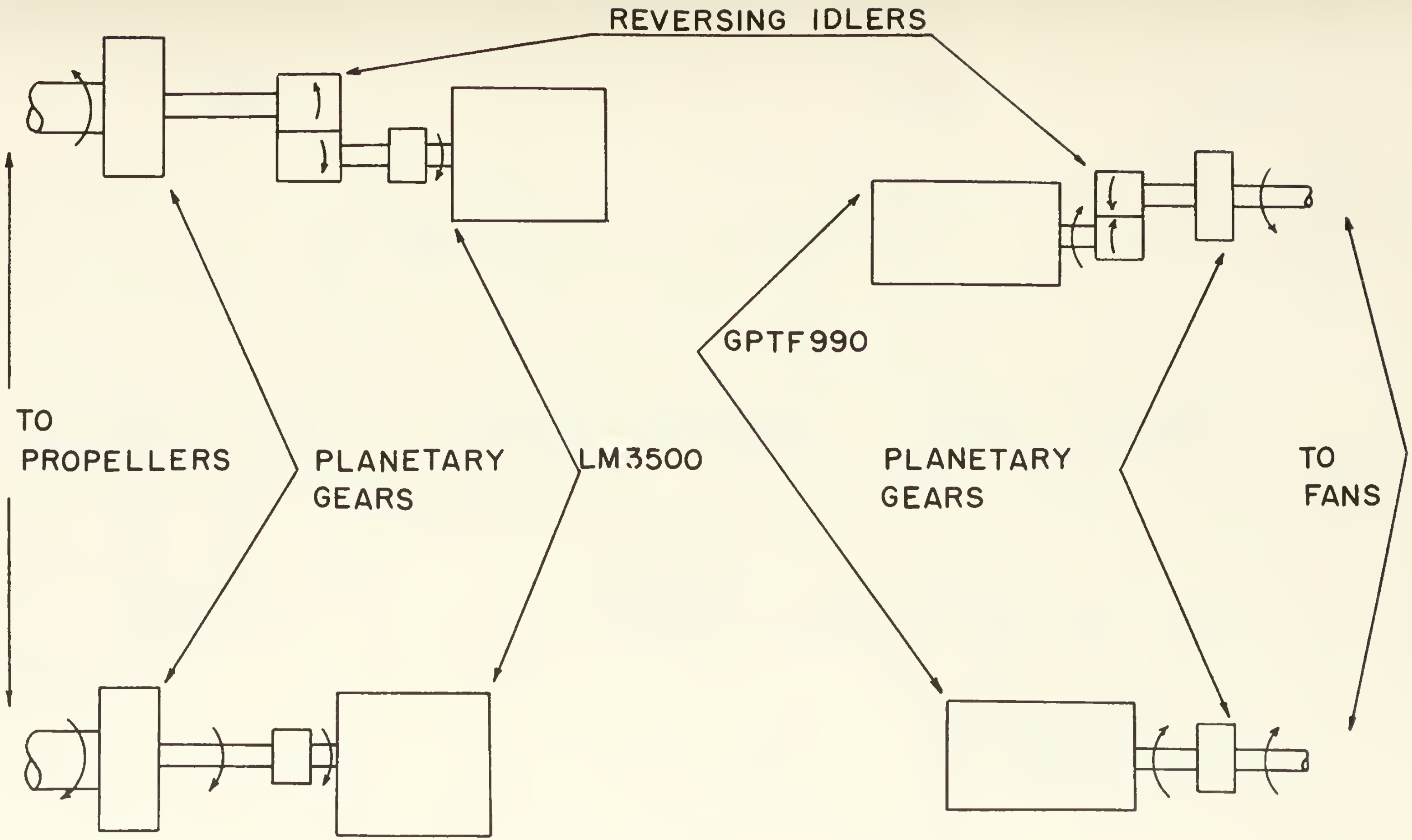


FIGURE 23 FOUR TURBINE SPLIT PLANT TRANSMISSION SCHEMATIC

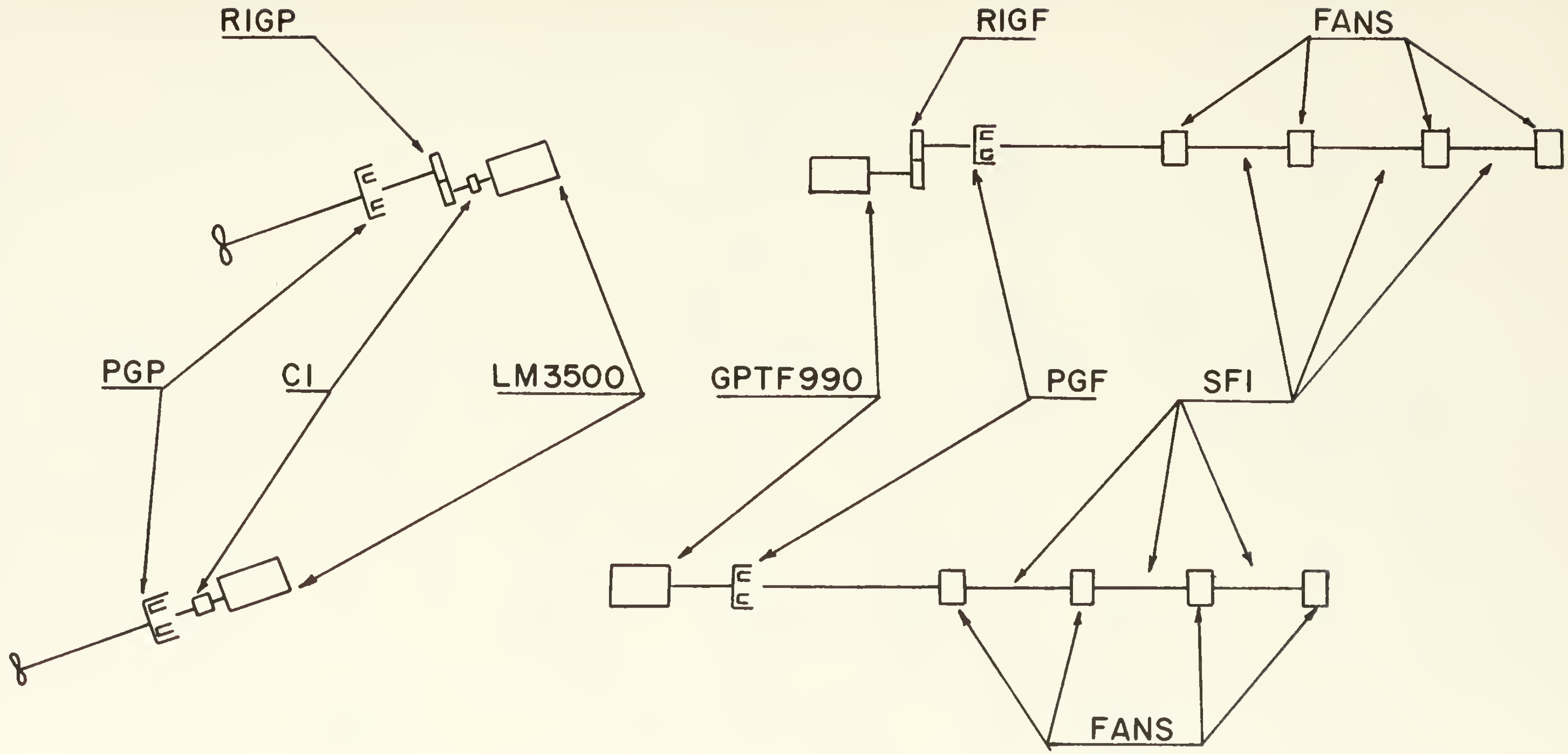
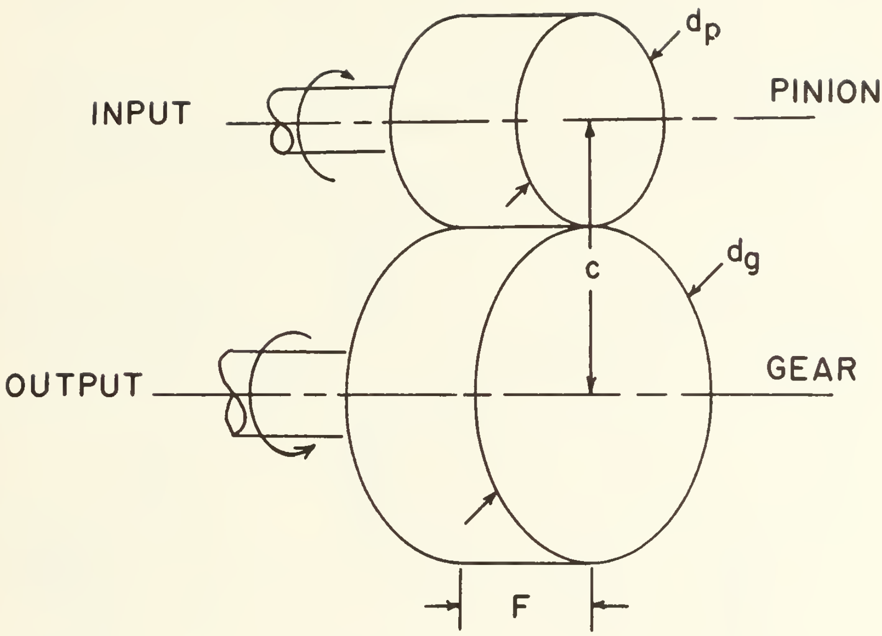


FIGURE 24 FOUR TURBINE SPLIT PLANT TRANSMISSION DIAGRAM



REDUCTION RATIO $m_g = \frac{d_g}{d_p}$

FIGURE 25 SINGLE REDUCTION GEAR NOMENCLATURE

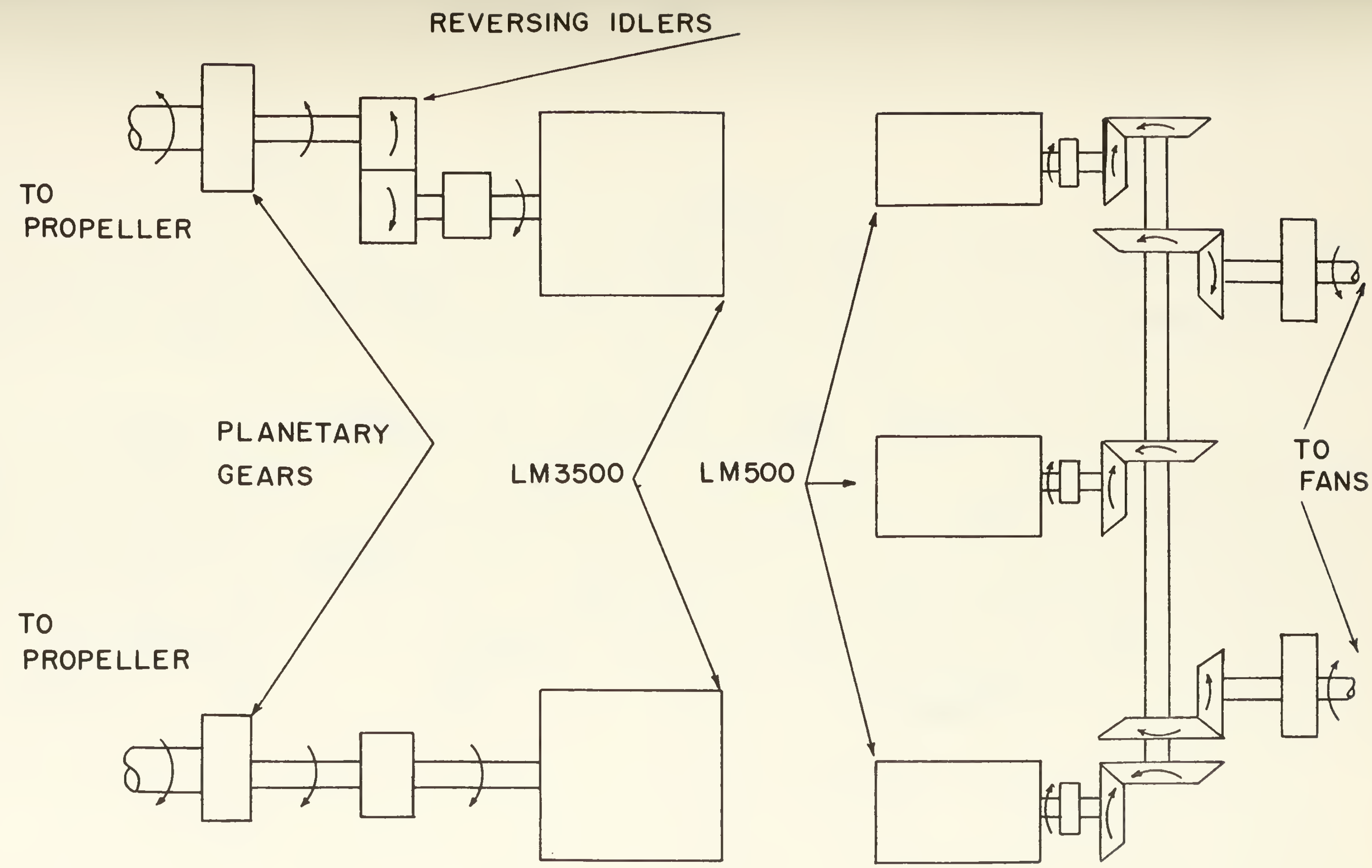


FIGURE 26 FIVE TURBINE SPLIT PLANT TRANSMISSION SCHEMATIC

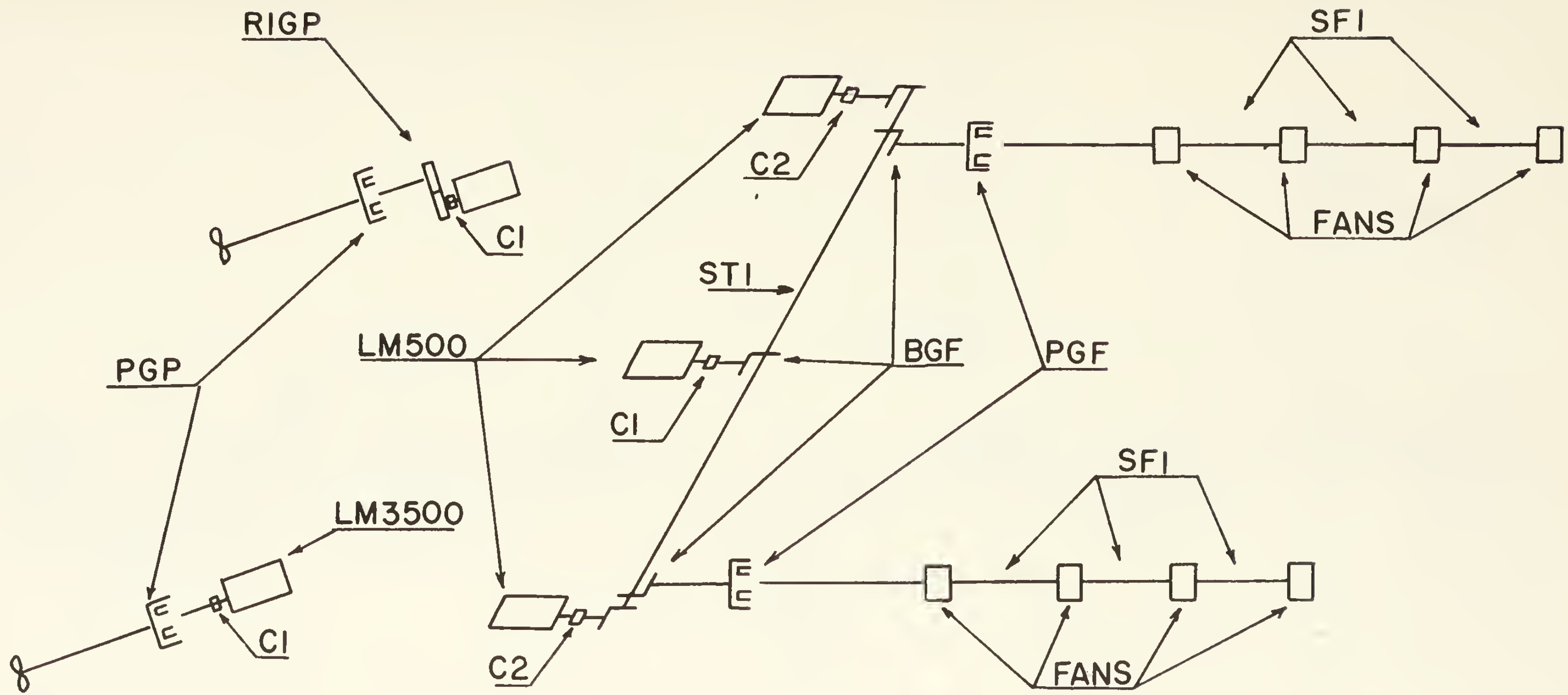


FIGURE 27 FIVE TURBINE SPLIT PLANT TRANSMISSION DIAGRAM

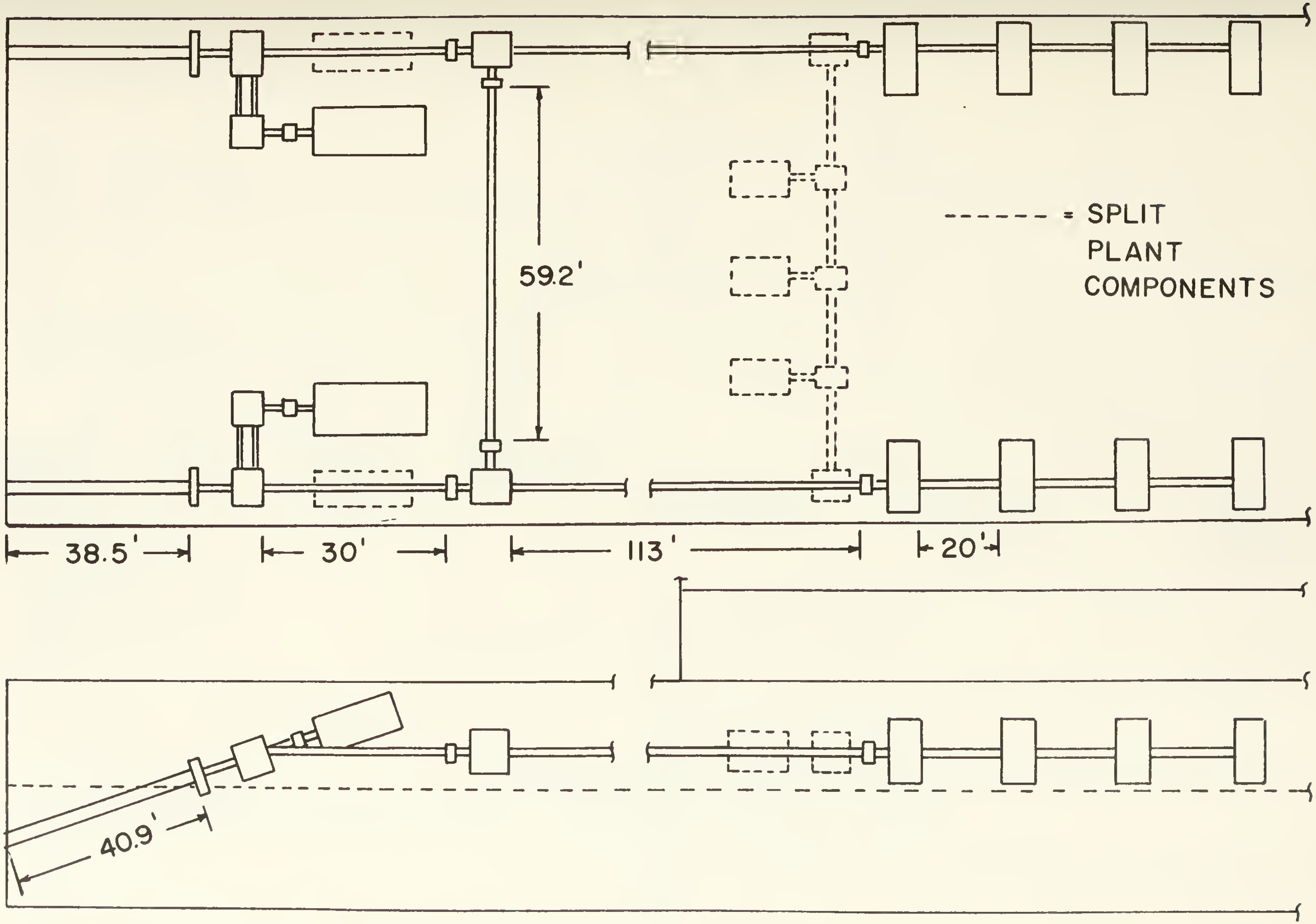


FIGURE 28. SES DIMENSIONAL PLANT LAYOUT

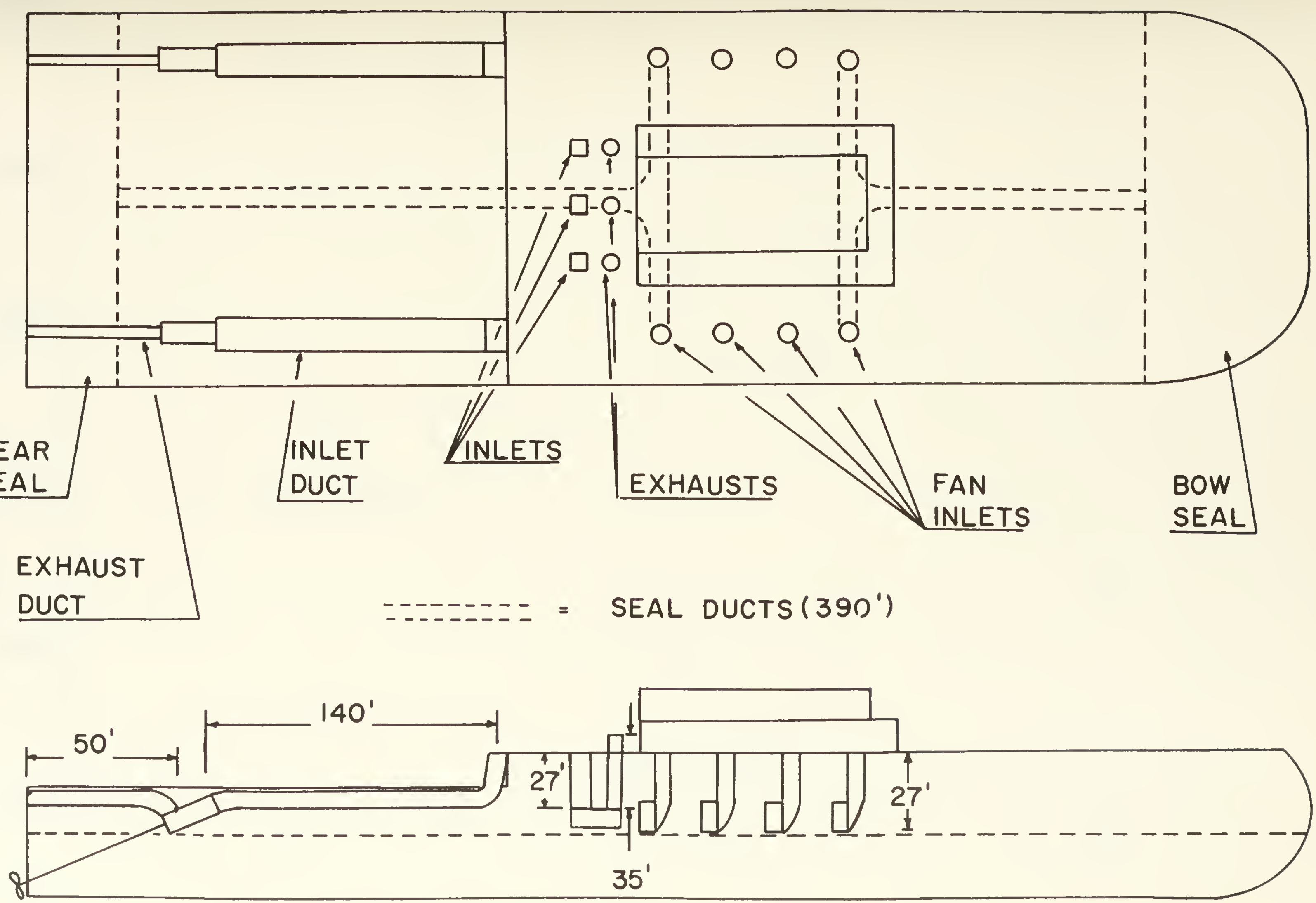


FIGURE 29 SES DUCTING DIAGRAM

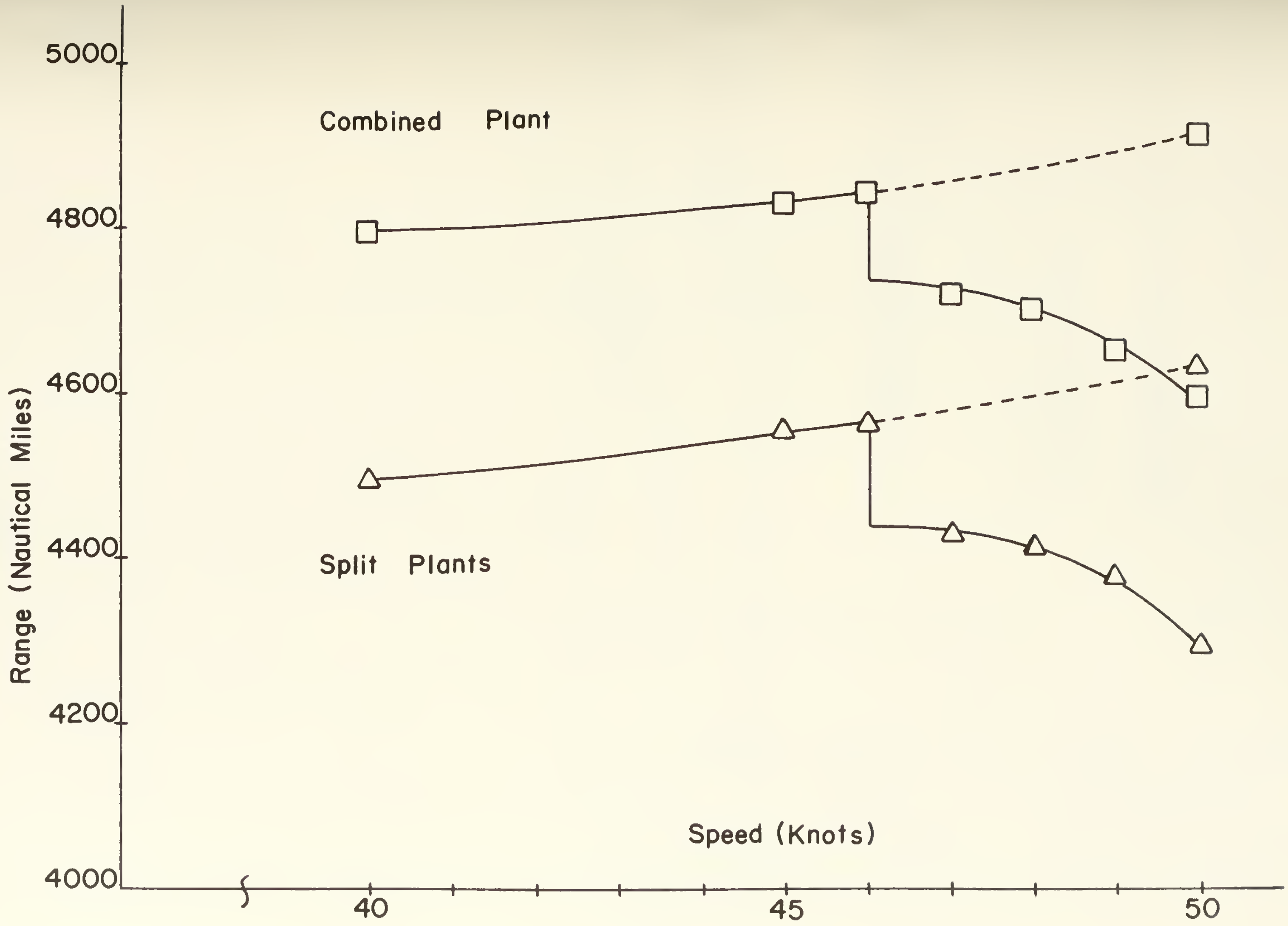


FIGURE 30 Range Integration Results

TABLE 1

5000 TON SES CHARACTERISTIC DIMENSIONS

Cushion Length, L_c	=	450	feet
Cushion Beam, B_c	=	69.23	feet
Sidewall (Sidehull) Beam, B_{SW} (B_{SH})	=	7.00	feet
Overall Beam, B_{OA}	=	83.23	feet
Sidewall Depth (Cushion Height), H_c	=	18.00	feet
Number of Decks, n_d	=	4	
Deck Height, H_d	=	9	feet
Vehicle Depth (Overall Height), H_{OA}	=	$n_d \times H_d + H_c$	
	=	$(4)(9) + 18$	
	=	54	feet
Cushion Pressure, P_c	=	325	lb/ft^2
Cushion Density, $\frac{\Delta}{(L_c B_c)^{3/2}}$	=	2.04	lb/ft^3

TABLE 2

SUMMARY OF ASSUMPTIONS USED IN THE ANALYSIS

ASSUMPTIONS	APPLIES TO: Split Plant Only	Combined Plant Only	Both Plant Types Only
1. The SES is operating at a constant 45 knots in sea state 3.			X
2. The propeller thrust and hence ship speed is controlled by varying the pitch of the partially submerged supercavitating controllable pitch (PSS-CRP) propellers not by changing propeller RPM.			X
3. The cushion air pressure is varied by changing the inlet geometry of the lift fans.			X
4. The lift fans are operated at a constant RPM.			X
5. The propulsors are operated at a constant RPM.			X
6. The load on the lift fan engines varies due to torque changes on the fan shafts.	X		
7. The load on the engines of the combined plant will vary due to torque changes transmitted by the propeller and fan shafts.		X	
8. As the SES consumes fuel it becomes lighter, thus requiring less cushion air pressure.			X
9. Reduced cushion pressure means a reduction in lift horsepower requirements which, in turn, results in less torque and therefore less load on the engines.			X
10. The weight of fuel plus propulsion and lift hardware will be a fixed stated percentage of the SES's gross weight.			X

TABLE 3

FEASIBLE PROPELLERS FOR THE 5000 TON SES

PROPELLER	SUBMERGENCE %	ANGLE OF		γ_o	J	K_T	K_Q	D ft	n RPS	T_p $\times 10^5$	Q_p $\times 10^5$	$(HP)_B$ $\times 10^4$
		Inclination(°)	Rake(°)									
NSRDC 4281	50	20	20	0.68	1.32	.045	.0125	16.3	4.53	1.301	5.894	3.029
NSRDC 4281	30	10	10	0.68	1.64	.025	.008	25.2	2.55	1.300	10.48	3.024
NSRDC 4281	30	20	20	0.70	1.00	.050	.0125	13.6	6.25	1.295	4.48	3.099
NSRDC 4407	30	20	19.5	0.68	1.46	.050	.0160	17.8	3.59	1.300	7.42	3.024

TABLE 4

FAN SIZE AND EFFICIENCY

r/r_{max}	$D_s = F(N_s)$
1.0	$D_s = 1.20 + 1.90\ (\frac{N_s}{60})^{-1.24}$
0.98	$D_s = 1.12 + 1.45\ (\frac{N_s}{60})^{-1.65}$
0.95	$D_s = 1.07 + 1.33\ (\frac{N_s}{60})^{-1.85}$
0.90	$D_s = 0.98 + 1.23\ (\frac{N_s}{60})^{-1.97}$
0.80	$D_s = 0.92 + 1.08\ (\frac{N_s}{60})^{-2.01}$

TABLE 5

RECOMMENDED TIP SPEED LIMITS FOR LIFT FANS

FAN TYPE	U_t (ft/sec)	LIMIT RATIONALE
Centrifugal & Mixed Flow	500	Conservative stress limit
	800	Ultimate noise limit
Axial	650	Conservative stress limit
	800	Ultimate noise limit

TABLE 6

SUMMARY OF 5000 TON SES FAN CHARACTERISTICS

P_c	=	325	lb^3/ft^2
Q_f	=	1951	ft^3/sec
Q_T	=	15,611	ft^3/sec
N_s	=	100	
$D_s/60$	=	1.57	
ψ	=	0.398	
N	=	1233.9	RPM
D	=	8.56	ft
L	=	6.86	ft
A	=	16.01	ft
W_f	=	2509	lbs
$(HP)_L$	=	14,079	H.P.
$_{max}$	=	0.82	
$_f$	=	0.78	
$_d$	=	0.90	
$_T$	=	0.98	

TABLE 7

MARINE GAS TURBINES

MANUFACTURER	MODEL	CONT. POWER RATING (hp)	SFC lb/HP-HR	W lbs	SPECIFIC WEIGHT lbs/HP	RPM	PRESS. RATIO	ENGINE VOL. (ft^3)
United Aircraft of Canada Ltd.	ST6J-70	489	0.67	350	0.7	2,200	6	
	ST6K-70	489	0.67	317	0.62	6,230	6	
	ST6J-77	529	0.663	379	0.69	2,200	6	
	ST6K-77	529	0.663	350	0.64	6,230	6	
	ST6L-77	633	0.618	306	0.47	33,000	6.3	
	FT4A-2	24,200	.50	14,200	.587	3,600	-	1,195.6
	FT4A-12	26,950	.51	14,950	.55	3,600	-	1,195.6
	FT4A-14	31,150	.51	14,600	.47	3,600	-	1,195.6
Avco Lycoming	AGT 450	450	.47	1,260	2.8	22,500	-	26.6
	AGT 750	750	.42	1,650	2.2	22,500	-	32.4
	AGT 1000	1,000	.42	2,000	2.0	22,500	-	38.2
	TF-12	1,090	0.64	920	0.84	18,000	6.1	
	TF-14	1,390	0.60	920	0.66	19,600	6.5	
	AGT 1500	1,500	.42	2,000	1.5	22,500	-	41.2
	TF-25	2,130	0.66	1,020	0.48	15,100	6.1	
	TF-35	2,675	0.57	1,090	0.41	15,350	6.5	
		3,000						
	TF-40	3,440	0.53	1,160	0.34	15,350	7.2	
	TF-50	4,300	.46	1,250	.29	15,400	-	40.0
Rolls Royce	Tyne RMIA	4,250	.49	20,000[1]	4.7	12,900	12.0	430.6
	Tyne RMIC	5,340	.47	20,000	3.7	13,970	12.5	430.6
	Protcus	3,600	.60	3,118	.867	11,600	7.3	131.8
	Olympus TM	21,000	.51	47,000	2.24	5,660	10.3	1101.3
Allison	501	3,780	.54	2,500	.66	13,820	-	100
Solar IHC	T-1000	1,200	.63	1,250	1.04	22,300	-	80.2

TABLE 7 (CONT'D)

MARINE GAS TURBINES

MANUFACTURER	MODEL	CONT. POWER RATING (hp)	SFC lb/HP-HR	W lbs	SPECIFIC WEIGHT lbs/HP	RPM	PRESS. RATIO	ENGINE VOL. (ft^3)
Pratt & Whitney	FT12	3,620	0.71	1,150	0.36	9,000	7	
	FT4C-2	30,600	0.46	16,100	0.53	3,600	12	
	FT93	40,000	0.37	16,500	0.41	3,600	18	
Garrett/AiResearch	GTP 831	475	.76	1,500	3.16	39,000	-	31.6
	IE 831-800	600	.64	1,700	2.83	41,730	11.0	49.8
	GTPF 990	5,900	.46	5,200[3]	.87	3,600	-	210
General Electric	LM500	4,500	0.46	1,180	0.26	6,500	14	
	LM1500	14,800	0.54	7,500	0.51	5,500	12	
	LM2500	22,500	0.4	10,600	0.47	3,600	15.45	
	LM3500[3]	34,950	0.38	13,000	0.37	3,600	21.76	
	LM5000[3]	46,000	0.36	19,750	0.43		31.3	

1) Free Power Turbine
2) With integral gearbox
3) Proposed

TABLE 8

SUMMARY OF 5000 TON SES GAS TURBINES

ENGINE	APPLICATION	PLANT TYPE	$(HP)_B$	SFC lb/HR	RPM	WEIGHT (lbs)	LENGTH	DIAMETER
FT 9	Propulsion & Lift	Combined	38,024	0.405	3600	16,500	22 ft-6"	7 ft-11"
LM 3500	Propulsion	Split 4&5 Turbine Plants	31,000	0.472	3600	13,000	25 ft-4"	8 ft- 4"
GPTF 990	Lift	Split 4 Turbine Plant	7,024	0.460	3600	6,125	8 ft-2"	4 ft
LM 500	Lift	Split 5 Turbine Plant	4,693	0.430	6500	1,180	27 ft	7 ft-11"

TABLE 9

SUMMARY OF REDUCTION GEAR WEIGHT AND SIZE ESTIMATION FORMULAS

GEAR DRIVE TYPE	LIMITATION	GEARBOX WEIGHT lb	GEAR SIZE in.	GEARBOX SIZE in.
Single reduction	None	$W=2.9\times10^4(Q/K)^{0.8}$ (full weight) $W=1.6\times10^4(Q/K)^{0.9}$ (light weight)	$d_p = 40(R/K)^{1/3}$ $d_g = d_p m_g$ $F = 2d_p$	$h= 1.3(d_p + d_g)$ $w= 1.3\ d_g$ $t= 1.5\ F$
Planetary reduction (epicyclic)	None	$W=0.95\times10^4(Q/K)$	$d_s = 40\ {\frac{1}{bk}\frac{P_s}{n_s}\frac{m_g}{m_g - 2}}^{1/3}$ $d_r = d_s(m_g - 1)$ $F = 2\ d_s$	$d= 1.5\ d_r$ $l= 3F$
Bevel	Maximum gear diameter 24 in.; special 35 in.	$W = 280\ Q^{0.8}$ single mesh $W = 220\ Q^{0.8}$ double mesh	$d_p = {\frac{1600}{1+m_g^2}}^{1/3} R^{0.32}, R \leq 1.5$ $d_p = {\frac{1400}{1+m_g^2}}^{1/3} R^{0.42}, R > 1.5$ $d_g = d_p m_g$	$d= 1.5\ d_g$, $h= 2d_p$ single mesh $l= 3d_{g1}$, $w= 1.5d_{g1}$, $h= 2d_{p1}$ double mesh first gearbox
Remarks	-	$Q = \frac{P}{n_p}\frac{(m_g + 1)^3}{m_g}$	$R = \frac{P}{n_p}\frac{m_g + 1}{bm_g}$	$l= 1.5(d_{g1}+d_{p2})$, $w= 1.5\ d_{g2}$, $h= 2d_{g2}$ double mesh second gearbox

TABLE 10

MAXIMUM NUMBER OF PLANETS
FOR A PLANETARY REDUCTION GEAR

Reduction Ratio, m_g	Number of planets(b)
$<$ 13.4	3
$<$ 6.3	4
$<$ 4.5	5
$<$ 3.7	6
$<$ 3.4	7
$<$ 3.1	8

TABLE 11

SUMMARY OF THE 5000 TON SES COMBINED PLANT TRANSMISSION SYSTEM

DESCRIPTION	NO.	H.P. X 10^3	RPM X 10^3	DIAMETER (in)	LENGTH (in)	UNIT WEIGHT (lb)	TOTAL WEIGHT (lb)
9.6:1 Propulsion Planetary Gear (PGP)	2	31.0	3.6	99.9	46.4	20,298	40,596
2.93:1 Fan Planetary Gear (PGF)	2	7.0	3.6	13.4	27.7	765	1,530
1:1 Bevel Gear (BG1) (port and starboard)	2	38.0	3.6	40.2	80.4	7,650	15,300
1:1 Bevel Gear (BG2) (allow $\frac{1}{2}$ weight of normal gearbox)	2	14.0	3.6	35.4	35.4	2,190	4,380
1:1 Bevel Gear (BG3)	2	31.0	3.6	36.9	73.8	6,500	13,000
1:1 Bevel Gear (BG4) (cross connect)	2	14.0	3.6	35.4	47.2	4,380	8,760
1:1 Bevel Gear (BG5) (allow $\frac{1}{2}$ weight of normal gearbox)	2	7.0	3.6	26.4	35.3	1,258	2,516
						Total Gear Weight	86,082

Transmission Nomenclature

PG(m) = Planetary Gear (F for fan, P for propulsion)
RIG(m) = Reversing Idler Gear (F for fan, P for propulsion)
BG(n) = Bevel Gear (number)
SI(n) = Shaft, interconnecting (number)
SP = Shaft, propulsion section
SF(n) = Sahft, fan section (number)
C(n) = Clutch (number)

TABLE 11 (CONT'D)

SUMMARY OF THE 5000 TON SES COMBINED PLANT TRANSMISSION SYSTEM

DESCRIPTION	NO.	H.P. X 10^3	RPM X 10^3	f	SIZE				WEIGHT (lb)				
					D_o (in)	L_{CR} (in)	L_o (in)	No. Elem.	W_{LS}	W_c	W_M	W_{elem}	W_{tot}
SP1	4	19.0	3.6	0.91	8.13	58.02	60.0	1	146	219	196.5	561.5	2246
SI1	2	14.0	3.6	0.90	7.13	54.2	360	6.6	116	161.8	161	438.8	5792
SI2	1	14.0	3.6	0.90	7.13	54.2	708	13	116	161.8	161	438.8	5704
SI3	2	7.0	3.6	0.88	5.38	46.9	1356	29	92	81	105	278	16,124
SF1	6	7.0	1.227	0.915	8.49	101.7	240	2.4	395	237	359.8	991.8	14,282
C1 - clutch to isolate turbines	2	38.0	3.6									2989	5978
C2 - clutch to isolate cross-connect	2	14.0	3.6									1103	2206
C3 - clutch to isolate propeller shafting	2	14.0	3.6									1103	2206

Total Weight of Shafting and Clutches 54,538

TABLE 12

SUMMARY OF THE 5000 TON SES 4 TURBINE SPLIT PLANT TRANSMISSION SYSTEM

DESCRIPTION	NO.	H.P. X 10^3	RPM X 10^3	DIAMETER (in)	LENGTH (in)	UNIT WEIGHT (lb)	TOTAL WEIGHT (lb)
9.6:1 Propulsion Planetary Gear (PGP)	2	31.0	3.6	99.9	46.4	20,298	40,596
2.93:1 Fan Planetary Gear (PGF)	2	7.0	3.6	13.4	27.7	765	1,530
1:1 Direction reversing idler gear-propulsion (RIGP)	1	31.0	3.6	36.8	12.9	2,688	2,688
1:1 Direction reversing idler gear-fans (RIGF)	1	7.0	3.6	22.4	7.8	705	705

Total Gear Weight 45,519

DESCRIPTION	NO.	H.P. X 10^3	RPM X 10^3	ξ	SIZE				WEIGHT (lb)				
					D_o (in)	L_{CR} (in)	L_o (in)	No. Elem.	W_{LS}	W_c	W_M	W_{elem}	W_{tot}
SF1 - fan shafts	6	7.0	1.227	0.915	8.5	101.7	240	2.4	395	237	360	992	14,285
C1 - clutches to isolate turbines	2	31.0	3.6									2442	4,884

Total Weight of Shafting and Clutches 19,169

TABLE 13

SUMMARY OF THE 5000 TON SES 5 TURBINE SPLIT PLANT TRANSMISSION SYSTEM

DESCRIPTION	NO.	H.P. X 10^3	RPM X 10^3	DIAMETER (in)	LENGTH (in)	UNIT WEIGHT (lb)	TOTAL WEIGHT (lb)
9.6:1 Propulsion Planetary Gear (PGP)	2	31.0	3.6	99.9	46.4	20,298	40,596
2.93:1 Fan Planetary Gear (PGF)	2	7.0	3.6	13.4	27.7	765	1,530
1:1 Direction reversing idler gear-propulsion (RIGP)	1	31.0	3.6	36.8	12.9	2,688	2,688
1:1 Bevel gears (BGF)	5	9.33	6.5	23.3	31.0	1,974	9,870

Total Gear Weight 54,684

DESCRIPTION	NO.	H.P. X 10^3	RPM X 10^3	ζ	SIZE				WEIGHT (lb)				
					D_o (in)	L_{CR} (in)	L_o (in)	No. Elem.	W_{LS}	W_c	W_M	W_{elem}	W_{tot}
ST1-shafting between turbines	1	9.33	6.5	0.86	4.7	32.3	708	21.9	40.6	59.7	62.9	163.2	3,574
SF1-shafting between fans	6	7.00	1.227									992	14,285
C1-clutches to isolate propulsion turbines	2	31.0	3.6									2442	4,884
C2-clutches to isolate fan turbines	3	4.667	6.5									203.6	611

Total Weight of Shafting and Clutches 23,354

TABLE 14

SUMMARY OF PROPELLER SHAFTING FORMULAS

8.2 $$Q = \frac{(63,025)SHP}{RPM}$$

Q = Torsional Shear Stress ; in-lb.

8.3 $Q'_{Navy} = 1.2Q$ (Navy shafting shall be designed for full power torque plus a 20% overload torque.)

8.4 $$S_s = \frac{Q'(d_o/2)}{\frac{\pi}{32}(d_o^4 - d_i^4)}$$

S_s = Shear Stress ; lb/in^2

d_o = outside diameter, in.

d_i = inner diameter, in.

8.5 $$S_c = \frac{T}{A} = \frac{(EHP)(550)}{V(1-t)(\pi/4)(d_o^2 - d_i^2)}$$

S_c = Compressive Stress ; lbs/in^2

A = cross sectional area, in^2

V = ship speed, ft/sec

$1-t$ = thrust deduction factor (assume $1-t = 1$)

$EHP = (SHP)(NPC)$

8.6 $$S_R = ((S_c^2) + (2S_s)^2)^{\frac{1}{2}}$$

S_R = Resultant Steady Stress ; lbs/in^2

8.7 $$M_g = W_p L_p$$

M_g = Gravity Moment ; in-lb

W_p = Propeller Weight ; lbs

TABLE 14 (CONT'D)

SUMMARY OF PROPELLER SHAFTING FORMULAS

8.7 Cont'd

L_p = Shaft Length (from last bearing support to propeller) ; in.

(Assume L_p = 60 in)

$M_R = 2M_g$

M_R = Resultant Moment ; in-lb

8.8 $$(S_B)_{alt} = \frac{M_R(d_o/2)}{\frac{\pi}{32}(d_o^4 - d_i^4)}$$

$(S_B)_{alt}$ = Alternating Bending Stress ; lbs/in^2

8.9 $(S_{TS}) = r(S_s)$; $0.02 \le r \le 0.12$, select $r = 0.12$

(S_{TS}) = Torsion Stress ; lbs/in^2

8.10 $(S_R)_{alt} = ((K_b S_b)^2 + (2K_T S_s)^2)^{\frac{1}{2}}$

$(S_R)_{alt}$ = Resultant Alternating Stress ; lbs/in^2

K_b, K_T = stress concentration factors (assume $K_b = K_T = 1.0$)

8.11 $$\frac{1}{FS} = \frac{\text{Resultant Steady Stress}}{YP} + \frac{\text{Resultant Alternating Stress}}{FL}$$

FS = Factor of Safety (assume FS = 2 for tail shaft)

YP = Yield Point

FL = Failure Level

TABLE 15a

SUMMARY OF GAS TURBINE INLET DUCTING WEIGHTS

TURBINE	Wt lb/ft^2	COMBUSTION AIR Mass Flow Rate lbm/sec	Volume Flow Rate ft^3/sec	1.1XVolume Flow Rate ft^3/sec	Velocity ft/sec	Duct x-sect Area ft^2	Duct Length ft	Weight lb
FT9	6	182	2355	2590	70	37	140	20,430
LM3500	6	164	2122	2334	70	33.3	140	19,386
GPTF990	6	151	1954	2149	70	30.7	27	3,191
LM500	4	137	1773	1950	70	27.9	27	2,280

TABLE 15b

SUMMARY OF GAS TURBINE EXHAUST DUCTING WEIGHTS

TURBINE	Wt lb/ft^2	Velocity ft/sec	Duct x-sect Area ft^2	Duct Length ft	Weight lb
FT9	7	150	15.7	50	4916
LM3500	7	150	15.6	50	4900
GPTF990	7	150	14.3	35	3284
LM500	5	150	13.0	35	2237

TABLE 16

COMPONENT SUMMARY OF THE COMBINED PLANT FOR THE 5000 TON SES

DESCRIPTION	NO.	H.P. X 10^3	RPM X 10^3	DIAMETER	LENGTH	CROSS-SECT AREA (ft^2)	LENGTH (ft)	UNIT WEIGHT (lb)	TOTAL WEIGHT (lb)
FT9 gas turbine	2	38.0	3.6	95 in.	270 in.			16,500	33,000
Gas turbine inlet ducting	2					37	140	10,215	20,430
Gas turbine exhaust ducting	2					15.7	50	2,458	4,916
PSSCRP - propeller	2	31.0	0.375	13.5 ft				19,689	39,378
Propeller shaft	2	31.0	0.375	20in-outer 13.3in-inner	40.97ft			13,756	27,512
Propeller shaft coupling	2	31.0	0.375					1,026	2,052
Lubrication and bearing supports	2							4,100	8,200
Fans	8	1.75	1.227	8.56 ft	6.84 ft			2,509	20,072
Fan inlet ducting	8			6.25 ft	27 ft			3,313	26,504
Seal ducting	2			6.25 ft	390 ft			7,657	15,315

Total Component Weight 197,379

TABLE 17

COMPONENT SUMMARY OF THE 4 TURBINE SPLIT PLANT FOR THE 5000 TON SES

DESCRIPTION	NO.	H.P. X 10^3	RPM X 10^3	DIAMETER	LENGTH	CROSS-SECT AREA (ft^2)	LENGTH (ft)	UNIT WEIGHT (lb)	TOTAL WEIGHT (lb)
LM3500 gas turbine	2	31.0	3.6	99.9 in	303 in			13,000	26,000
GPTF990 gas turbine	2	7.0	3.6	48 in	96 in			6,125	12,250
LM3500 inlet ducting	2					33.3	140	19,386	38,772
LM3500 exhaust ducting	2					15.6	50	4,900	9,800
GPTF990 inlet ducting	2					30.7	27	3,191	6,382
GPTF990 exhaust ducting	2					14.3	35	3,284	6,568
PSSCRP propellers	2	31.0	0.375	13.5 ft				19,689	39,378
Propeller shaft	2	31.0	0.375	20in-outer 13.3in-inner	40.97 ft			13,756	27,512
Propeller shaft coupling	2	31.0	0.375					1,026	2,052
Lubrication and bearing support	2							4,100	8,200
Fans	8	1.75	1.227	8.56 ft	6.84 ft			2,509	20,072
Fan inlet ducting	8			6.25 ft	27 ft			3,313	26,504
Seal ducting	2			6.25 ft	390 ft			7,657	15,315

Total Component Weight 238,805

TABLE 18

COMPONENT SUMMARY OF THE 5 TURBINE SPLIT PLANT FOR THE 5000 TON SES

DESCRIPTION	NO.	H.P. X 10^3	RPM X 10^3	DIAMETER	LENGTH	CROSS-SECT AREA (ft^2)	LENGTH (ft)	UNIT WEIGHT (lb)	TOTAL WEIGHT (lb)
LM3500 gas turbine	2	31.0	3.6	99.9 in	303 in			13,000	26,000
LM500 gas turbine	3	4.667	6.5	108 in	66 in			1,219	3,657
LM3500 inlet ducting	2					33.3	140	19,386	38,772
LM3500 exhaust ducting	2					15.6	50	4,900	9,800
LM500 inlet ducting	3					27.9	27	2,280	4,560
LM500 exhaust ducting	3					13.0	35	2,237	6,711
PSSCRP propellers	2	31.0	0.375	13.5 ft				19,689	39,378
Propeller shaft	2	31.0	0.375	20in-outer 13.3in-inner	40.97 ft			13,756	27,512
Propeller shaft coupling	2	31.0	0.375					1,026	2,052
Lubrication and bearing supports	2							4,100	8,200
Fans	8	1.75	1.227	8.56 ft	6.84 ft			2,509	20,072
Fan inlet ducting	8			6.25 ft	27 ft			3,313	26,504
Seal ducting	2			6.25 ft	390 ft			7,657	15,315

Total Component Weight 228,533

TABLE 19

SUMMARY OF ALTERNATIVE POWER PLANT SYSTEM WEIGHTS

	COMBINED PLANT	FOUR TURBINE SPLIT PLANT	FIVE TURBINE SPLIT PLANT
Reduction Gears (lbs/tons)	86,082 / 38.4	45,519 / 20.3	54,684 / 24.4
Transmission shafting and clutches (lbs/tons)	54,538 / 24.4	19,169 / 8.6	23,354 / 10.4
Components (lbs/tons)	197,379 / 88.1	238,805 /106.6	228,533 /102
Total (lbs/tons)	337,999 /151	303,493 /135.5	306,571 /136.8

	COMBINED PLANT	FOUR TURBINE SPLIT PLANT	FIVE TURBINE SPLIT PLANT
Total Hardware Weight (tons)	151	135.5	136.8
Fuel Weight (25% Fuel Fraction)(tons)	1235.8	1251.3	1250
Total System Weight (tons)	1386.8	1386.8	1386.8

Note: The five turbine split plant is used as the baseline; i.e. the five turbine split plant carries a 25% fuel fraction.

TABLE 20

COMBINED PLANT RANGE CALCULATIONS

(25% FUEL FRACTION, 40 KNOTS, SS-3)

FUEL %	FUEL WT. (ton)	P_C (lb/ft^2)	Q_R (ft^3/sec)	$(HP)_C$	$(HP)_P$	$(HP)_{TOT}$	$(SFC)_C$	$(SFC)_P$	$(SFC)_T$	RANGE NM
100	1235.8	325	15,610.8	14,079	46,462	60,541	0.3925	0.3925	0.3925	932.1
80	988.6	307.2	15,172	12,925		59,337	0.3944	0.3944	0.3944	946.4
60	741.5	289.5	14,733.5	11,836.3		58,298	0.3961	0.3961	0.3961	959.1
40	494.3	271.7	14,273.4	10,761.7		57,224	0.3979	0.3979	0.3979	972.8
20	247.2	253.9	13,797.9	9,721.6		56,184	0.3997	0.3997	0.3997	986.3
0	0									

Total 4797

TABLE 21

4 TURBINE SPLIT PLANT RANGE CALCULATIONS

(25% FUEL FRACTION, 40 KNOTS, SS-3)

FUEL %	FUEL WT. (tons)	P_C (lb/ft^2)	Q_R (ft^3/sec)	$(HP)_C$	$(HP)_P$	$(HP)_{TOT}$	$(SFC)_C$	$(SFC)_P$	$(SFC)_T$	RANGE NM
100	1251.3	325	15,610.8	14,079	46,462	60,541	0.4600	0.4164	0.4265	868.6
80	1001.0	307	15,172	12,925.7		59,387.7	0.4674		0.4275	883.4
60	750.8	289	14,721	11,806		58,268	0.4770		0.4287	897.8
40	500.5	271	14,255	10,720		57,182	0.4870		0.4296	913.0
20	250.3	253	13,773.5	9,670		56,132	0.500		0.4308	927.4
0	0									

Total 4490

TABLE 22

5 TURBINE SPLIT PLANT RANGE CALCULATIONS

(25% FUEL FRACTION, 40 KNOTS, SS-3)

FUEL %	FUEL WT. (tons)	P_C (lb/ft^2)	Q_R (ft^3/sec)	$(HP)_C$	$(HP)_P$	$(HP)_{TOT}$	$(SFC)_C$	$(SFC)_P$	$(SFC)_T$	RANGE NM
100	1250	325	15,610.8	14,079	46,462	60.541	0.460	0.4164	0.4265	867.5
80	1000	307	15,172.3	12,925.7		59,388	0.4674		0.4275	882.3
60	750	289	14,720.8	11,805.7		58,268	0.4766		0.4286	886.9
40	500	271	14,255	10,720		57,182	0.4872		0.4297	911.7
20	250	253	13,773.5	9,670		56,132	0.5000		0.4308	926.3
0	0									

Total 4485

TABLE 23

COMBINED PLANT RANGE CALCULATIONS

(25% FUEL FRACTION, 45 KNOTS, SS-3)

FUEL %	FUEL WT. (tons)	P_C (lb/ft^2)	Q_R (ft^3/sec)	$(HP)_C$	$(HP)_P$	$(HP)_{TOT}$	$(SFC)_C$	$(SFC)_P$	$(SFC)_T$	RANGE NM
100	1235.8	325	15,610.8	14,079	55,556	69,635	0.3802	0.3802	0.3802	941.1
80	988.6	307.2	15,172	12,925		68,431	0.3816	0.3816	0.3816	954.2
60	741.5	289.5	14,733.5	11,836.3		67,392	0.3829	0.3829	0.3829	965.7
40	494.3	271.7	14,273.4	10,761.7		66,318	0.3843	0.3843	0.3843	977.7
20	247.2	253.9	13,797.9	9,721.6		65,278	0.3857	0.3857	0.3857	989.8
0	0									

Total 4828

TABLE 24

4 TURBINE SPLIT PLANT RANGE CALCULATIONS

(25% FUEL FRACTION, 45 KNOTS, SS-3)

FUEL %	FUEL WT. (tons)	P_C (lb/ft^2)	Q_R (ft^3/sec)	$(HP)_C$	$(HP)_P$	$(HP)_{TOT}$	$(SFC)_C$	$(SFC)_P$	$(SFC)_T$	RANGE NM
100	1251.3	325	15,610.8	14,079	55,556	69,635	0.4600	0.3980	0.4105	882.7
80	1001.0	307	15,172	12,925.7		68,509	0.4674		0.4109	896.3
60	750.8	289	14,721	11,806		67,362	0.4770		0.4118	909.5
40	500.5	271	14,255	10,720		66,276	0.4870		0.4124	923.1
20	250.3	253	13,773.5	9,670		65,226	0.5000		0.4131	936.4
0	0									

Total 4548

TABLE 25

5 TURBINE SPLIT PLANT RANGE CALCULATIONS

(25% FUEL FRACTION, 45 KNOTS, SS-3)

FUEL %	FUEL WT. (tons)	P_C (lb/ft^2)	Q_R (ft^3/sec)	$(HP)_C$	$(HP)_P$	$(HP)_{TOT}$	$(SFC)_C$	$(SFC)_P$	$(SFC)_T$	RANGE NM
100	1250	325	15,610.8	14,079	55,556	69,635	0.4600	0.3980	0.4105	881.6
80	1000	307	15,172.3	12,925.7	↓	68,482	0.4674	↓	0.4111	895.1
60	750	289	14,720.8	11,805.7		67,362	0.4766		0.4118	908.5
40	500	271	14,255	10,720		66,276	0.4872		0.4124	922.0
20	250	253	13,773.5	9,670		65,226	0.5000		0.4131	935.2
0	0									

Total 4542

TABLE 26

COMBINED PLANT RANGE CALCULATIONS

(25% FUEL FRACTION, 46 KNOTS, SS-3)

FUEL %	FUEL WT. (tons)	P_C (lb/ft^2)	Q_R (ft^3/sec)	$(HP)_C$	$(HP)_P$	$(HP)_{TOT}$	$(SFC)_C$	$(SFC)_P$	$(SFC)_T$	RANGE NM
100	1235.8	325	15,610.8	14,079	57,281	71,360	0.3782	0.3782	0.3782	943.8
80	988.6	307.2	15,172.3	12,925	↓	70,206	0.3795	0.3795	0.3795	956.0
60	741.5	289.5	14,733.5	11,836.3	↓	69,117	0.3808	0.3808	0.3808	967.8
40	494.3	271.7	14,273.4	10,761.7	↓	68,043	0.3821	0.3821	0.3821	979.7
20	247.2	253.9	13,797.9	9,721.6	↓	67,003	0.3834	0.3834	0.3834	991.5
0	0									

Total 4838

TABLE 27

4 TURBINE SPLIT PLANT RANGE CALCULATIONS

(25% FUEL FRACTION, 46 KNOTS, SS-3)

FUEL %	FUEL WT. (tons)	P_C (lb/ft^2)	Q_R (ft^3/sec)	$(HP)_C$	$(HP)_P$	$(HP)_{TOT}$	$(SFC)_C$	$(SFC)_P$	$(SFC)_T$	RANGE NM
100	1251.3	325	15,610.8	14,079	57,281	71,360	0.4600	0.3957	0.4084	885.0
80	1001.0	307	15,172	12,925.7	↓	70,207	0.4674	↓	0.4089	898.4
60	750.8	289	14,721	11,806	↓	69,087	0.4770	↓	0.4096	911.4
40	500.5	271	14,255	10,720	↓	68,001	0.4870	↓	0.4101	924.8
20	250.3	253	13,773.5	9,670	↓	66,951	0.5000	↓	0.4108	937.7
0	0									

Total 4557

TABLE 28

5 TURBINE SPLIT PLANT RANGE CALCULATIONS

(25% FUEL FRACTION, 46 KNOTS, SS-3)

FUEL %	FUEL WT. (tons)	P_C (lb/ft^2)	Q_R (ft^3/sec)	$(HP)_C$	$(HP)_P$	$(HP)_{TOT}$	$(SFC)_C$	$(SFC)_P$	$(SFC)_T$	RANGE NM
100	1250	325	15,610.8	14,079	57,281	71,360	0.4600	0.3957	0.4084	883.9
80	1000	307	15,172	12,925.7	↓	70,206	0.4674	↓	0.4089	897.3
60	750	289	14,721	11,805.7	↓	69,117	0.4770	↓	0.4096	909.9
40	500	271	14,255	10,720	↓	68,043	0.4870	↓	0.4101	923.1
20	250	253	13,773.5	9,670	↓	67,003	0.5000	↓	0.4108	935.9
0	0									

Total 4550

TABLE 29

COMBINED PLANT RANGE CALCULATIONS

(25% FUEL FRACTION, 50 KNOTS, SS-3, STATIC LIFT CASE)

FUEL %	FUEL WT. (tons)	P_C (lb/ft^2)	Q_R (ft^3/sec)	$(HP)_C$	$(HP)_P$	$(HP)_{TOT}$	$(SFC)_C$	$(SFC)_P$	$(SFC)_T$	RANGE NM
100	1235.8	325	15,610.8	14,079	63,558	77,637	0.3718	0.3718	0.3718	959.1
80	988.6	307.2	15,172	12,925	↓	76,433	0.3730	0.3730	0.3730	971.1
60	741.5	289.5	14,733.5	11,836.3	↓	75,394	0.3740	0.3740	0.3740	981.9
40	494.3	271.7	14,273.4	10,761.7	↓	74,320	0.3751	0.3751	0.3751	993.1
20	247.2	253.9	13,797.9	9,721.6	↓	73,280	0.3761	0.3761	0.3761	1004.6
0	0									

Total 4910

TABLE 30

4 TURBINE SPLIT PLANT RANGE CALCULATIONS

(25% FUEL FRACTION, 50 KNOTS, SS-3, STATIC LIFT CASE)

FUEL %	FUEL WT. (tons)	P_C (lb/ft^2)	Q_R (ft^3/sec)	$(HP)_C$	$(HP)_P$	$(HP)_{TOT}$	$(SFC)_C$	$(SFC)_P$	$(SFC)_T$	RANGE NM
100	1251.3	325	15,610.8	14,079	63,558	77,637	0.4600	0.3869	0.4002	902.2
80	1001.0	307	15,172	12,925.7		76,483.7	0.4674		0.4005	915.2
60	750.8	289	14,721	11,806		75,364	0.4765		0.4009	927.8
40	500.5	271	14,255	10,720		74,278	0.4872		0.4014	940.3
20	250.3	253	13,733.5	9,670		73,228	0.4998		0.4018	952.8
0	0									

Total 4638

TABLE 31

5 TURBINE SPLIT PLANT RANGE CALCULATIONS

(25% FUEL FRACTION, 50 KNOTS, SS-3, STATIC LIFT CASE)

FUEL %	FUEL WT. (tons)	P_C (lb/ft^2)	Q_R (ft^3/sec)	$(HP)_C$	$(HP)_P$	$(HP)_{TOT}$	$(SFC)_C$	$(SFC)_P$	$(SFC)_T$	RANGE NM
100	1250	325	15,610.8	14,079	63,558	77,637	0.4600	0.3869	0.4002	901.2
80	1000	307	15,172.3	12,925.7	↓	76,484	0.4674	↓	0.4005	914.1
60	750	289	14,720.8	11,805.7	↓	75,364	0.4766	↓	0.4009	926.8
40	500	271	14,255	10,720	↓	74,278	0.4872	↓	0.4014	939.1
20	250	253	13,773.5	9,670	↓	73,228	0.4998	↓	0.4018	951.6
0	0									

Total 4633

TABLE 32

COMBINED PLANT RANGE CALCULATIONS

(25% FUEL FRACTION, 47 KNOTS, SS-3, WAVE PUMPING CASE)

FUEL %	FUEL WT. (tons)	P_C (lb/ft^2)	Q_R (ft^3/sec)	$(HP)_C$	$(HP)_P$	$(HP)_{TOT}$	$(SFC)_C$	$(SFC)_P$	$(SFC)_T$	RANGE NM
100	1235.8	325	15,937.5	14,374	59,023	73,397	0.3760	0.3760	0.3760	943.0
80	988.6									
60	741.5									
40	494.3									
20	247.2									
0	0									

Total 4715

TABLE 33

4 TURBINE SPLIT PLANT RANGE CALCULATIONS

(25% FUEL FRACTION, 47 KNOTS, SS-3, WAVE PUMPING CASE)

FUEL %	FUEL WT. (tons)	P_C (lb/ft^2)	Q_R (ft^3/sec)	$(HP)_C$	$(HP)_P$	$(HP)_{TOT}$	$(SFC)_C$	$(SFC)_P$	$(SFC)_T$	RANGE NM
100	1251.3	325	15,937.5	14,374	59,023	73,397	0.4577	0.3931	0.4057	884.9
80	1001.0									
60	750.8									
40	500.5									
20	250.3									
0	0									

Total 4424

TABLE 34

5 TURBINE SPLIT PLANT RANGE CALCULATIONS

(25% FUEL FRACTION, 47 KNOTS, SS-3, WAVE PUMPING CASE)

FUEL %	FUEL WT. (tons)	P_C (lb/ft^2)	Q_R (ft^3/sec)	$(HP)_C$	$(HP)_P$	$(HP)_{TOT}$	$(SFC)_C$	$(SFC)_P$	$(SFC)_T$	RANGE NM
100	1250	325	15,937.5	14,374	59,023	73,397	0.4577	0.3931	0.4057	883.9
80	1000									
60	750									
40	500									
20	250									
0	0									

Total 4419

TABLE 35

COMBINED PLANT RANGE CALCULATIONS

(25% FUEL FRACTION, 48 KNOTS, SS-3, WAVE PUMPING CASE)

FUEL %	FUEL WT. (tons)	P_C (lb/ft^2)	Q_R (ft^3/sec)	$(HP)_C$	$(HP)_P$	$(HP)_{TOT}$	$(SFC)_C$	$(SFC)_P$	$(SFC)_T$	RANGE NM
100	1235.8	325	16,276.6	14,679	61,015	75,694	0.3737	0.3737	0.3737	939.7
80	988.6									
60	741.5									
40	494.3									
20	247.2									
0	0									

Total 4699

TABLE 36

4 TURBINE SPLIT PLANT RANGE CALCULATIONS

(25% FUEL FRACTION, 48 KNOTS, SS-3, WAVE PUMPING CASE)

FUEL %	FUEL WT. (tons)	P_C (lb/ft^2)	Q_R (ft^3/sec)	$(HP)_C$	$(HP)_P$	$(HP)_{TOT}$	$(SFC)_C$	$(SFC)_P$	$(SFC)_T$	RANGE NM
100	1251.3	325	16,276.6	14,679	61,015	75,694	0.4559	0.3902	0.4029	882.5
80	1001.0									
60	750.8									
40	500.5									
20	250.3									
0	0									

Total 4411.5

TABLE 37

5 TURBINE SPLIT PLANT RANGE CALCULATIONS

(25% FUEL FRACTION, 48 KNOTS, SS-3, WAVE PUMPING CASE)

FUEL %	FUEL WT. (tons)	P_C (lb/ft^2)	Q_R (ft^3/sec)	$(HP)_C$	$(HP)_P$	$(HP)_{TOT}$	$(SFC)_C$	$(SFC)_P$	$(SFC)_T$	RANGE NM
100	1250	325	16,276.6	14,679	61,015	75,694	0.4559	0.3902	0.4029	881.4
80	1000									
60	750									
40	500									
20	250									
0	0									

Total 4407

TABLE 38

COMBINED PLANT RANGE CALCULATIONS

(25% FUEL FRACTION, 49 KNOTS, SS-3, WAVE PUMPING CASE)

FUEL %	FUEL WT. (tons)	P_C (lb/ft²)	Q_R (ft³/sec)	$(HP)_C$	$(HP)_P$	$(HP)_{TOT}$	$(SFC)_C$	$(SFC)_P$	$(SFC)_T$	RANGE NM
100	1235.8	325	16,615.7	14,985	63,603	78,588	0.3709	0.3709	0.3709	930.8
80	988.6	↓	↓	↓	↓	↓	↓	↓	↓	↓
60	741.5									
40	494.3									
20	247.2									
0	0									

Total 4654

TABLE 39

4 TURBINE SPLIT PLANT RANGE CALCULATIONS

(25% FUEL FRACTION, 49 KNOTS, SS-3, WAVE PUMPING CASE)

FUEL %	FUEL WT. (tons)	P_C (lb/ft²)	Q_R (ft³/sec)	$(HP)_C$	$(HP)_P$	$(HP)_{TOT}$	$(SFC)_C$	$(SFC)_P$	$(SFC)_T$	RANGE NM
100	1251.3	325	16,615.7	14,985	63,603	78,588	0.4542	0.3868	0.3996	874.7
80	1001.0									
60	750.8									
40	500.5									
20	250.3									
0	0									

Total 4373

TABLE 40

5 TURBINE SPLIT PLANT RANGE CALCULATIONS

(25% FUEL FRACTION, 49 KNOTS, SS-3, WAVE PUMPING CASE)

FUEL %	FUEL WT. (tons)	P_C (lb/ft^2)	Q_R (ft^3/sec)	$(HP)_C$	$(HP)_P$	$(HP)_{TOT}$	$(SFC)_C$	$(SFC)_P$	$(SFC)_T$	RANGE NM
100	1250	325	16,615.7	14,985	63,603	78,588	0.4542	0.3868	0.3996	873.8
80	1000									
60	750									
40	500									
20	250									
0	0									

Total 4369

TABLE 41

COMBINED PLANT RANGE CALCULATIONS

(25% FUEL FRACTION, 50 KNOTS, SS-3, WAVE PUMPING CASE)

FUEL %	FUEL WT. (tons)	P_C (lb/ft^2)	Q_R (ft^3/sec)	$(HP)_C$	$(HP)_P$	$(HP)_{TOT}$	$(SFC)_C$	$(SFC)_P$	$(SFC)_T$	RANGE NM
100	1235.8	325	16,954.8	15,291	66,558	81,848	0.3680	0.3680	0.3680	919.2
80	988.6									
60	741.5									
40	494.3									
20	247.2									
0	0									

Total 4596

TABLE 42

4 TURBINE SPLIT PLANT RANGE CALCULATIONS

(25% FUEL FRACTION, 50 KNOTS, SS-3, WAVE PUMPING CASE)

FUEL %	FUEL WT. (tons)	P_C (lb/ft^2)	Q_R (ft^3/sec)	$(HP)_C$	$(HP)_P$	$(HP)_{TOT}$	$(SFC)_C$	$(SFC)_P$	$(SFC)_T$	RANGE NM
100	1251.3	325	16,954.8	15,291	66,558	81,848	0.4525	0.3869	0.3992	858.1
80	1001.0									
60	750.8									
40	500.5									
20	250.3									
0	0									

Total 4290

TABLE 43

5 TURBINE SPLIT PLANT RANGE CALCULATIONS

(25% FUEL FRACTION, 50 KNOTS, SS-3, WAVE PUMPING CASE)

FUEL %	FUEL WT. (tons)	P_C (lb/ft^2)	Q_R (ft^3/sec)	$(HP)_C$	$(HP)_P$	$(HP)_{TOT}$	$(SFC)_C$	$(SFC)_P$	$(SFC)_T$	RANGE NM
100	1250	325	16,954.8	15,291	66,558	81,848	0.4525	0.3869	0.3992	856.9
80	1000									
60	750									
40	500									
20	250									
0	0									

Total 4285

APPENDIX A

SAMPLE CALCULATIONS OF RANGE INTEGRATION RESULTS

The purpose of these sample calculations is to illustrate how the values summarized in Tables 20-43 were generated.

1. Cushion Pressure (P_c). In the static lift case, the lift system must provide the difference between the gross weight of the vessel and the bouyant force (F_B) acting on the sidehulls :

$$F_B = \Delta_{FL} - \frac{P_c L_c B_c}{2240} \qquad (A.1)$$

where F_B = bouyant force acting on the sidehull, tons

Δ_{FL} = full load displacement, tons

P_c = cushion pressure, lb/ft^2

L_c = cushion length, ft.

B_c = cushion beam, ft.

Using the values for the 5000 ton SES :

$$F_B = 480 \text{ tons}$$

The cushion pressure required may be calculated from :

$$P_c = \frac{(W_{SC} - W_{FB})\, 2240}{B_c L_c} \qquad (A.2)$$

where P_c = cushion pressure, lb/ft^2

W_{SC} = ship weight supported by cushion pressure, W_{SC} = 5000 - 480 = 4520 tons

W_{FB} = weight of fuel burned off, tons

Using the values for the 5000 ton SES :

$$P_c = \frac{(4520 - W_{FB})(2240)}{(69.23)(450)} = 7.1902 \times 10^{-2}(4520 - W_{FB}) \quad (A.3)$$

2. Cushion Air Flow Rate (Q).

a. The cushion air flow rate required by the static lift cases (Tables 20-31), is calculated using equation (5.6a) :

$$Q_{SL} = S_g D_C \left(\frac{2 P_c}{a} \right)^{\frac{1}{2}} \quad (5.6a)$$

Using the values of h_g = o.333 ft and D_C = 0.65, the static lift flow requirement is :

$$Q_{SL} = 8.6593 \times 10^2 (P_c)^{\frac{1}{2}} \quad (A.4)$$

b. The cushion air flow rate required by the wave pumping cases is calculated from :

$$Q_{WP} = B_c h_w V \quad (5.7a)$$

$$Q_{WP} = (339.1)(V \text{ knots}) \quad (A.5)$$

Equation (A.5) was used to calculate the cushion air flow rate entered in Tables 32-43. (Applies to speeds 46 knots.)

3. Cushion Horsepower (HP). The cushion horsepower is calculated using equation (5.13) :

$$(HP)_C = \frac{(P_c + \Delta P_c) Q}{550 \gamma_f \gamma_D \gamma_T} \quad (5.13)$$

Using the values given in Chapter 5 :

$$(HP)_C = 2.775 \times 10^{-3} P_c Q \quad (A.6)$$

4. Propulsive Horsepower $(HP)_P = (HP)_{BR-}$. The propulsive horsepower required by the SES is calculated using equation (4.1) and Figure 7 :

$$(HP)_P = \frac{(D_{TOT})_{RW} V}{(550)(NPC)(\gamma_T)} \quad (4.1)$$

Using values of NPC = 0.66, and $\gamma_T = 0.97$:

$$(HP)_P = 2.840 \times 10^{-2} (D_{TOT})_{RW} V \quad (A.7)$$

The values of smooth water drag $(D_{TOT})_{SW}$ at the speed being examined are read from Figure 7. The rough water drag $(D_{TOT})_{RW}$ is calculated from :

$$(D_{TOT})_{RW} = 1.1 (D_{TOT})_{SW} \quad (3.13)$$

5. Total Horsepower $(HP)_{TOT}$. The total horsepower required is the sum of the lift horsepower and propulsive horsepower :

$$(HP)_{TOT} = (HP)_C + (HP)_T \quad (A.8)$$

6. Range Calculations.

a. This calculation will determine the range achieved by the combined plant while it is consuming fuel at the 80% level and at a speed of 40 knots (Table 20). Since the same prime mover provides both lift and propulsion power, only a single SFC needs to be calculated :

$$f' = (0.491 n' + 0.316) t' + 0.196 n' - 0.004 \quad (6.1)$$

At 40 knots : $n' = 1.000$

$$t' = \frac{59,337 \text{ H.P.}}{80,000 \text{ H.P.}} = 0.7417$$

$$f' = 0.7906$$

$$(SFC)_{NEW} = \frac{(SFC)_{DESIGN}(HP)_{DESIGN} f'}{(HP)_{REQUIRED}}$$

$$= \frac{(0.37)(80,000)(0.7906)}{(59,337)} = 0.3944 \frac{\text{lb}}{\text{HP-HR}}$$

$$(SFC)_{TOTAL} = \frac{(HP)_P(SFC)_P + (HP)_C(SFC)_C}{(HP)_P + (HP)_C} \quad (A.9)$$

where $(SFC)_{TOTAL}$ = specific fuel consumption for the entire plant, lbs/(HP-HR).

$(HP)_P$ = propulsive horsepower, H.P.

$(HP)_C$ = cushion horsepower, H.P.

$(SFC)_P$ = specific fuel consumption, propulsive turbines, lbs/(HP-HR).

$(SFC)_C$ = specific fuel consumption, cushion turbines, lbs/(HP-HR).

For the combined plant $(SFC)_C = (SFC)_P$.

$$\therefore \quad (SFC)_{TOTAL} = 0.3944 \text{ lbs/(HP-HR)}$$

The range equation is :

$$\text{Range} = \frac{W_{FUEL}\ V\ 2240}{(SFC)_{TOTAL}(HP)_{TOTAL}} \tag{A.10}$$

where Range = range in nautical miles

W_{FUEL} = weight of fuel consumed, tons

V = ship speed, knots.

For the combined plant at 40 knots :

$$\text{Range} = \frac{(247.2)(40)(2240)}{(0.3944)(46{,}462 + 12{,}925)} = 946.4 \text{ NM}$$

b. This calculation will determine the range achieved by the 4 turbine split plant consuming fuel at the 80% level and at a speed of 40 knots (Table 21). This case differs from the combined plant since the specific fuel consumption for the propulsion and lift system will be different :

<u>Cushion</u> $n' = 1.000$

$$t' = \frac{12{,}925.7}{14{,}070} = 0.9181$$

$$f' = 0.9329$$

$$(SFC)_C = \frac{(0.46)(14,079)(0.9329)}{(12,925.7)} = 0.4674 \frac{lb}{HP\text{-}HR}$$

<u>Propulsion</u> $n' = 1.000$

$$t' = \frac{46,462}{69,900} = 0.6647$$

$$f' = 0.7284$$

$$(SFC)_P = \frac{(0.38)(69,900)(0.7284)}{(46,462)} = 0.4164 \frac{lb}{HP\text{-}HR}$$

$$(SFC)_{TOTAL} = \frac{(0.4164)(46,462) + (0.4674)(12,925.7)}{(46,462) + (12,925.7)} = 0.4275 \frac{lb}{HP\text{-}HR}$$

$$\text{Range} = \frac{(250.3)(40)(2240)}{(0.4275)(46,462 + 12,925.7)} = 883.4 \text{ NM}$$

c. This calculation will determine the range achieved by the 5 turbine split plant consuming fuel at the 80% fuel level and at a speed of 40 knots (Table 22). Since this plant uses the same propulsion turbines as the 4 turbine plant and since the gas turbines for the cushion have the same SFC, the values of $(SFC)_P$ and $(SFC)_C$ and $(SFC)_{TOTAL}$ will be the same for both plants :

<u>Cushion</u> $n' = 1.000$

$$t' = \frac{12,925.7}{14,079} = 0.9181$$

$$f' = 0.9329$$

$$(SFC)_C = \frac{(0.46)(14,079)(0.9329)}{(12,925.7)} = 0.4674 \frac{lb}{HP\text{-}HR}$$

<u>Propulsion</u> $n' = 1.000$

$$t' = \frac{46,462}{69,900} = 0.6647$$

$$f' = 0.7284$$

$$(SFC)_P = \frac{(0.38)(69,900)(0.7284)}{(46,462)} = 0.4164 \frac{lb}{HP\text{-}HR}$$

$$(SFC)_{TOTAL} = \frac{(0.4164)(46,462) + (0.4674)(12,925.7)}{(46,462 + 12,925.7)} = 0.4275 \frac{lb}{HP\text{-}HR}$$

$$\text{Range} = \frac{(250.3)(40)(2240)}{(0.4275)(46,462 + 12,925.7)} = 882.3 \text{ NM}$$

d. This calculation will determine the range achieved by the combined plant at the 80% fuel level and at a speed of 47 knots (Table 32). The cushion pressure does not vary as the fuel level decreases for the wave pumping case. Therefore, the cushion pressure remains constant even though fuel is being consumed. Since the values of $(HP)_C$, $(HP)_P$, $(SFC)_C$, $(SFC)_P$ all remain constant at each speed level, the range at each fuel level for a specific speed is the same :

$$n' = 1.000$$

$$t' = \frac{73,397}{80,000} = 0.9175$$

$$f' = 0.9324$$

$$(SFC)_{NEW} = \frac{(SFC)_{DESIGN}\ (HP)_{DESIGN}\ f'}{(HP)_{REQUIRED}}$$

$$= \frac{(0.37)(80,000)(0.9175)}{73,397} = 0.3760 \frac{lb}{HP\text{-}HR}$$

$$\text{Range} = \frac{(247.2)(47)(2240)}{(0.3760)(73,397)} = 943.0 \text{ NM}$$

e. This calculation will determine the range achieved by the 4 turbine split plant at the 80% fuel level and at a speed of 47 knots (Table 33).

Cushion

$$n' = 1.000$$

$$t' = \frac{14,374}{14,079} = 1.021$$

$$f' = 1.0159$$

$$(SFC)_C = \frac{(0.46)(14,079)(1.0159)}{(14,374)} = 0.4577 \frac{lb}{HP\text{-}HR}$$

Propulsion $n' = 1.000$

$$t' = \frac{59,023}{69,900} = 0.8444$$

$$f' = 0.8734$$

$$(SFC)_P = \frac{(0.38)(69,900)(0.8734)}{(59,023)} = 0.3931 \frac{\text{lb}}{\text{HP-HR}}$$

$$(SFC)_{TOTAL} = \frac{(0.4577)(14,374) + (0.3931)(59,023)}{(14,374 + 59,023)} = 0.4057 \frac{\text{lb}}{\text{HP-HR}}$$

$$\text{Range} = \frac{(250.3)(47)(2240)}{(0.4057)(73,397)} = 884.9 \text{ NM}$$

f. This calculation will determine the range achieved by the 5 turbine split plant at the 80% fuel level and at a speed of 47 knots (Table 34).

Cushion $n' = 1.000$

$$t' = \frac{14,374}{14,079} = 1.021$$

$$f' = 1.0159$$

$$(SFC)_C = \frac{(0.46)(14,079)(1.0159)}{(14,374)} = 0.4577 \frac{\text{lb}}{\text{HP-HR}}$$

Propulsion $n' = 1.000$

$$t' = \frac{59,023}{69,900} = 0.8444$$

$$f' = 0.8734$$

$$(SFC)_P = \frac{(0.38)(69,900)(0.8734)}{(59,023)} = 0.3931 \frac{\text{lb}}{\text{HP-HR}}$$

$$(SFC)_{TOTAL} = \frac{(0.4577)(14,374) + (0.3931)(59,023)}{(14,374 + 59,023)} = 0.4057 \frac{\text{lb}}{\text{HP-HR}}$$

$$\text{Range} = \frac{(250)(47)(2240)}{(0.4057)(14,374 + 59,023)} = 883.9 \text{ NM}$$

Thesis 170583
H4865 Hergenreter
A comparison of a split and combined plant on a 5000 ton surface effect ship (SES).